

EMPIRICAL DESIGN

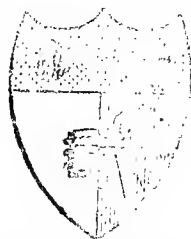
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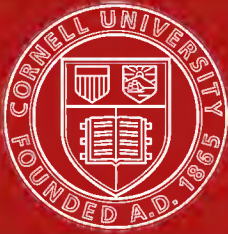
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EMPIRICAL DESIGN

BY

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PREFACE.

This book has been planned especially to meet the needs of the second year students in the Department of Machine Design of Cornell University. It is hoped, however, that the material is of a nature and the arrangement such as to meet, in a fair degree, the needs of other technical schools and colleges for a course in design to follow the elementary Mechanical Drawing and Descriptive Geometry before the student has had the necessary preparation in Mechanics for a course in theoretical design.

It is intended to give in a convenient form such tables, formulas and curves for the empirical proportioning of machine parts as are necessary in a brief course in Empirical Design, and such instruction as to the methods of their derivation and use that the student, upon the completion of the course, may be able to use material of this nature understandingly and to derive new material if he so desires. The data have been collected at different times and have been in use in teaching this course in Cornell University for a considerable period. The purpose and manner of using the various machine parts is explained in detail for the benefit of those students whose previous training has left them wholly unfamiliar with machinery. In the descriptions and explanations a fair knowledge of the principles of Mechanical Drawing, Descriptive Geometry and Analytical Geometry has been assumed.

No attempt has been made to intrude upon the field of the engineer's handbook but the proportions have been carefully compiled with the intent that any here given shall be reliable within the limits of good design, and some of the more commonly used mathematical tables have been added with a view to making it a desirable book for reference in the more advanced classes in Machine Design.

Acknowledgment is due to Professors E. H. Wood and C. D. Albert and to Mr. L. J. Bradford for suggestions and criticisms. The author is indebted to Professor Wood for the use of the proportions derived by him for many of the machine parts described, and several manufacturers have responded freely to requests for information.

June, 1915.

L. D. H.

CORRECTIONS FOR EMPIRICAL DESIGN

BY L. D. HAYES

- Page 13, line 7, for " $1\frac{7}{16}$ D" read " $1\frac{5}{16}$ D."
- " 20, " 8, for "countersunk" read "flat."
- " 37, " 20, to the equation for dimension F add "but not less than $2l-G$."
- " 47, Fig. 36 (e), invert.
- " 51, line 6, for "V" read "Y."
- " 58, " 18, add "F is the diameter of the bolts which sustain the bracket."
- " 70, Fig. 54, add "(a)" and "(b)" beneath the left-hand and right-hand portions respectively.
- " 87, line 10, for "page 91" read "page 92."
- " 87, " 19, in the parentheses add "in tees and crosses and one-half in laterals."
- " 99, top right-hand corner, for "Cotangents" read "Tangents."
- " 104, line 25, for "24" read "22."

CHAPTER I.

EMPIRICAL DESIGN.

1. **Definition.**—When a machine part has been proportioned from experience obtained in making other similar machine parts and without any direct application of the theory and principles of rational machine design it is said to have been designed empirically. Before the principles and laws which make modern design a fairly exact science had been discovered all design was of an empirical nature, differing little from guess work at first but becoming more and more exact as the result of increasing experience, and the elimination of such designs as were found to be too weak or too expensive in either material or labor.

2. **Empirical Methods in Modern Design.**—With the discovery of the laws governing the design of a machine part empirical methods of design were usually superseded. Sometimes this involved considerable change in the earlier designs but more often there was little change. It would seem, then, that continued study of these principles and laws must entirely displace empirical methods of design. There are, however, two classes of parts in the design of which empirical methods are likely to continue, e. g., those parts of such complex form that it is very difficult, if not impossible, to discover and apply the principles involved, and that large class of parts in use in so many sizes of similar proportions that, although rational methods may have been used in the design of the extreme sizes, the intermediate sizes are much more cheaply designed by empirical means based upon the proportions for the extreme sizes. The discussion of the first of these two classes is beyond the scope of this book but an attempt will be made to discuss some of the parts falling within this second class and to study the empirical methods used.

3. **Method of Application.**—The machine parts to which empirical methods will be applied belong to a large class of parts in such general use under such similar conditions that they are now made in a number of standard sizes having the same general form and

proportions and purchasable in the open market at a cost much less than that at which they could be designed and built especially for each case. Empirical methods are also often applied by manufacturers in the design of parts which they have occasion to make in a number of sizes even though those parts are never placed upon the market except in conjunction with some complete machine.

Any of these machine parts of modern origin have usually had their dimensions for a large and a small size determined by the principles of rational machine design, while the dimensions for parts of older origin may have been determined empirically. In either case the corresponding dimensions for the intermediate sizes, and sometimes for a small range beyond the selected extreme sizes, are determined according to some chosen law of variation. This may usually be done most conveniently by graphical methods.

To make a graphical determination of this kind the nominal sizes (by nominal size is meant that dimension which gives name to the size, as the outside diameter in the case of a handwheel), for the range through which it is desired to construct the part, are laid off to any convenient scale of abscissas and ordinates erected at these points. On the ordinates corresponding to the two sizes, usually near the limits of the desired range, for which all the dimensions are known (having been previously determined either by rational or by empirical methods) the values of these dimensions are laid off to scale. This scale should be so chosen that the dimensions may be read easily to as small a fraction of an inch as it is desirable to work in manufacturing the part. What the value of this fraction shall be is a matter for the judgment of the draftsman. His decision must be largely influenced by the size of the part, whether it is to be left in the rough or to be machined to size and, if it is to fit some other part, the nature of the fit required. Its value is rarely smaller than a sixteenth of an inch in unfinished castings, unless they are very small, and may often be as great as a fourth inch or a half inch in large castings. Having assumed the law of variations for one of these dimensions a curve following that law is drawn through the two points already determined for that dimension. The intersections of this curve with each of the remaining ordinates gives the value of this dimension for the corresponding sizes of the part to the same scale as that to which the known values were laid off.

As an example, in the handwheel, Fig. 1, let it be required to find the sizes of shafts (dimension B) suitable to use with a series of

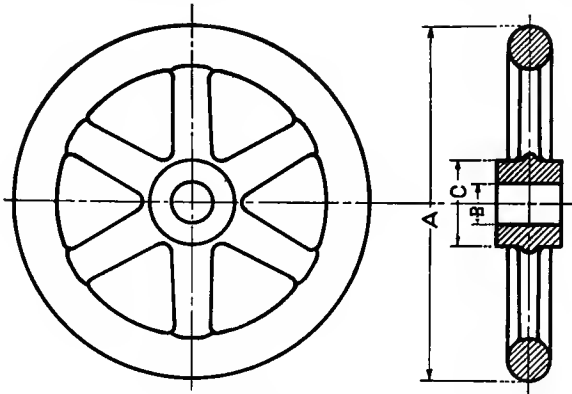


FIG. 1

nominal sizes (dimension A) when these dimensions are known to be $\frac{5}{8}$ " and $1\frac{1}{4}$ " for the 6" and 16" wheels respectively, and the variation assumed to be by direct proportion. On the base line, Fig. 2, lay off the nominal sizes and on the ordinates corresponding to the 6" and 16" sizes lay off to scale $\frac{5}{8}$ " and $1\frac{1}{4}$ " locating points

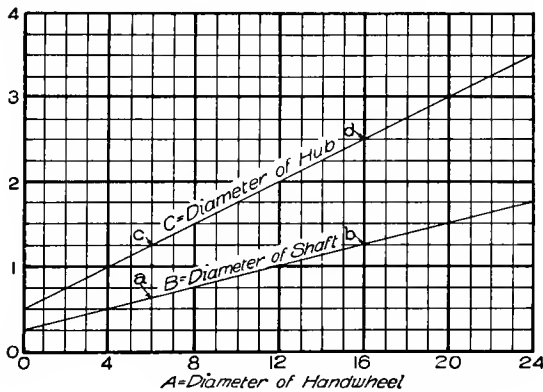


FIG. 2

a and b respectively. The straight line through these points represents the law of variation and by its intersection with the

several ordinates determines the required shaft diameters, as 1" for the 12" wheel. In the same way the line through points *c* and *d* determines the corresponding values for the diameters of hubs for these handwheels.

These curves may be left in their present form for the values to be read off when needed, the equations of the curves may be written and the values for the dimensions ascertained by substitutions in these equations, or the values for the dimensions for all sizes that it is desirable to manufacture may be read off from the curve at once and tabulated for use. Of these three methods the latter is the most convenient for general drafting room use while the first two have the advantage of showing clearly the relation between the several dimensions of the machine part.

4. Empirical Equations.—The equations have an advantage when compactness is a desirable quality and they may also be readily combined to show the relation between any two dimensions, whereas the curves refer all dimensions to the nominal size. For these reasons the equation method will be used quite largely in this book.

Applying the slope form, $y = mx + b$, of the equation for a straight line, to the curves of Fig. 2, and using the symbols of Fig. 1, gives

$$B = \frac{1}{16} A + \frac{1}{4}'' \quad \dots \dots \dots (1)$$

$$\text{and } C = \frac{1}{8} A + \frac{1}{2}'' \quad \dots \dots \dots (2).$$

As the diameter of the hub is more naturally a function of the diameter of the shaft than of the outside diameter of the handwheel, equations (1) and (2) may be combined to eliminate *A* giving

$$C = 2B \quad \dots \dots \dots (3).$$

When the variation desired for the dimensions of intermediate sizes does not conform to the straight line construction given above the careful design of one or two additional sizes will readily locate a suitable curve on the graphical construction, but it may be necessary to resort to some of the less simple methods of mathematics or to the use of logarithmic cross-section paper to obtain an equation for that curve. As these methods of determination permit considerable variation in the values taken for the dimensions it is often sufficiently accurate and much easier to approximate the

curve by two straight lines as illustrated in Fig. 3, where the curve B represents the selected law of variation for the thickness of babbitt in a certain type of bearing, in proportion to the diameter,

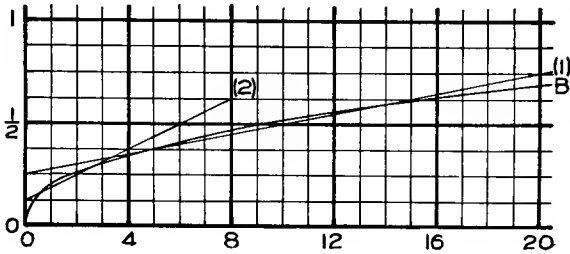


FIG. 3

A, of the shaft. The straight line (1) whose equation is $\frac{1}{40} A + \frac{1}{4}$ ", conforms closely to the curve for diameters from 4" to 16", which includes the sizes most commonly used, while the straight line (2), whose equation is $\frac{1}{16} A + \frac{1}{8}$ ", conforms well for the smaller diameters. We may, then, write

$B = \frac{1}{40} A + \frac{1}{4}$ " but not to exceed $\frac{1}{16} A + \frac{1}{8}$ " as fairly representative of the desired variation for the babbitt thickness for bearings of this type.

Many of the dimensions of a machine part are made up in a rational manner by adding together several other dimensions, the values of any or all of which may be of purely empirical origin.

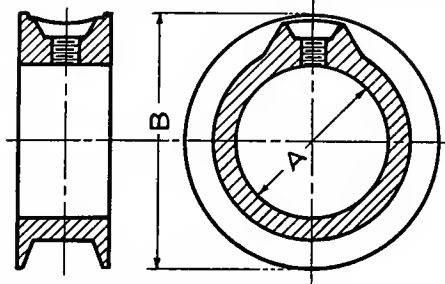


FIG. 4

An illustration of this occurs in the diameter, B, for the flange which protects the head of the set screw in the safety collar, shown

in Fig. 4. It is made up by adding to the diameter, A , of the bore for the shaft, twice the over all length of the set screw. This latter is, however, made up of the length, L , of the screw, measured from the tip of the point to the under side of the head, plus the thickness of the head which, in this particular form, is empirically taken to be one-half the diameter. We may, then, write

$$\begin{aligned} B &= A + 2(L + \tfrac{1}{2}d) \\ &= A + 2L + d. \end{aligned}$$

CHAPTER II.

SCREW FASTENINGS.

5. **General Forms.**—Screw fastenings are in use in so many ways that, besides the general standard forms which are common to all lines, there have been developed numerous forms which are standard for special classes of work and are carried in stock in regular sizes by the manufacturers. In this book the description will be confined to a few of those common forms which are carried in stock not only by the manufacturers but by most dealers in hardware or engineering supplies.

The proportions of these fastenings are empirical and, for some of them, the products of all makers do not agree in every particular, but they are in so close general agreement that for the same stock form most of them are interchangeable. Such fastenings are called for in the bill of material only, and no drawings made. If, however, any variation is made from the proportions for the stock form it becomes a "special" and a detail drawing must be made. Specials greatly increase the cost and should be avoided wherever it is possible to use a standard form.

The standard screw fastenings are divided into two general classes, namely, bolts and screws. In some of their forms the characteristics of the two so overlap that there is no clear distinction in their nomenclature. It is, perhaps, simplest for the student to learn the class name in conjunction with the form name for each case as he meets it.

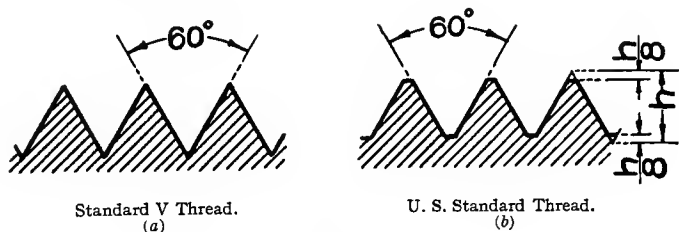


FIG. 5.

6. **Forms of Threads.**—While a number of different forms of screw threads have been devised, the full V, Fig. 5(a), and the United States Standard (also known as the "Sellers" and as the "Franklin Institute"), Fig. 5(b), are used practically to the exclusion of all others for screw fastenings in the United States. The United States Standard form, being only three-fourths as deep and with no sharp angle at the root of the threads, does not weaken the bolt nearly as much nor is it as easily mutilated as the full V thread. For these reasons it is generally preferred and is usually supplied by makers or dealers unless otherwise specified.

7. **Bolts.**—The usual form for a bolt has a head at one end with a nut which turns upon the threaded portion at the other end. Either the head or the nut may be omitted or replaced, in some of the forms. Bolts are used to hold members together in two distinct ways. They may be used to draw them together by tension between the head and nut or they may be passed through them as a pin or dowel, to prevent their sliding over each other, the nut and head serving to keep the bolt in place. In the former case it is not necessary that the holes should be finished or fit the bolt but in the latter case the bolt and holes must be machined to fit closely. In either case there should be a good bearing surface under head and nut and in the better class of work such surfaces should be machined. Wherever the nature of the work will not permit of a suitable bearing surface on the piece itself washers should be used. For further information regarding the use of washers see Art. 9, page 18.

The types of bolts which we shall discuss are illustrated in Fig. 6 and the numerical values of their dimensions, for diameters up to $3\frac{1}{2}$ ", are given in Table I. The relations between these dimensions are expressed in the following empirical equations.

$$T. P. I. = \frac{1}{0.24 \sqrt{D} + 0.625 - 0.175}$$

$$d = D - \frac{1.29904}{T. P. I.}$$

$$F = 1\frac{1}{2} D + \frac{1}{8}'' \text{ rough, } \frac{1}{16}'' \text{ less when finished.}$$

$$C = 1.155 F.$$

$$C' = 1.414 F.$$

$$T = D.$$

$$T' = \frac{3}{4} D \text{ to } D.$$

The length of a bolt, L , is measured between the points indicated in the figure, while the length of the perfect thread on the portion threaded to receive a nut is $S = 1\frac{1}{2} D$; and to enter a tapped hole in a machine member is $Z = \frac{7}{8} S = 1\frac{5}{16} D$. The hole into which this threaded portion, Z , enters should have a depth of perfect thread, $X = 1\frac{1}{8} S = 1\frac{11}{16} D$. Forms such as the machine bolt and stud bolt are frequently designated as "through bolts" to distin-

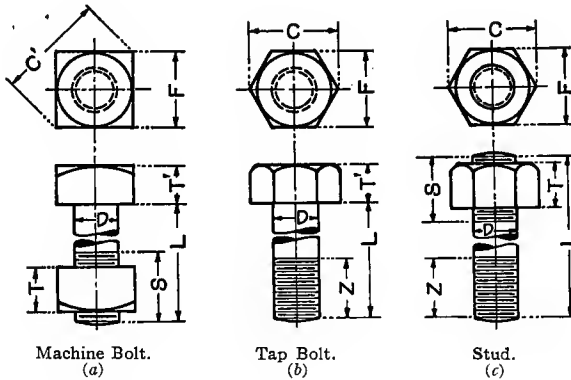


FIG. 6.

guish them from the tap bolt and stud which are threaded into a machine member.

Bolts with hexagon nuts cost about 10% more than those with square nuts. When both nuts and heads are hexagonal the cost is about 20% more than for square heads and nuts. Because of their greater cost bolts having hexagonal heads and nuts are used only when space is lacking, either to allow the corners of the nut to pass or to permit sufficient wrench movement for a square nut; or where a more finished appearance is desirable.

Machine Bolts.—The stock form of machine bolts, Fig. 6(a), have United States Standard thread with standard hexagonal or square head and standard hexagonal or square nut. Lengths vary by $\frac{1}{2}$ " from $1\frac{1}{2}$ " to 8", and by 1" from 8" to 30"; diameters from $\frac{1}{4}$ "

TABLE I.

U. S. STANDARD SCREW THREADS, NUTS AND BOLT HEADS.

Diameter of Body of Bolt.	Number of Threads per Inch.	Diameter at Root of Threads.	Area at Root of Threads.	Distance across Flats.	Diagonal of Head or Nut.		Thickness of Nut.
					Hexagonal.	Square.	
D	T.P.I.	d	a	F	C	C'	T
$\frac{1}{8}$	20	0.185	0.027	$\frac{1}{8}$	$\frac{9}{16}$	$\frac{23}{32}$	$\frac{1}{4}$
$\frac{1}{16}$	18	0.240	0.045	$\frac{1}{8}$	$\frac{11}{16}$	$\frac{3}{8}$	$\frac{5}{16}$
$\frac{3}{16}$	16	0.294	0.068	$\frac{1}{8}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{3}{8}$
$\frac{1}{2}$	14	0.344	0.093	$\frac{1}{8}$	$\frac{15}{16}$	$1\frac{1}{2}$	$\frac{7}{8}$
$\frac{5}{8}$	13	0.400	0.126	$\frac{1}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{8}$
$\frac{3}{4}$	12	0.454	0.162	$\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{3}{4}$	$1\frac{1}{4}$
$\frac{7}{8}$	11	0.507	0.202	$1\frac{1}{16}$	$1\frac{7}{8}$	$2\frac{1}{2}$	$1\frac{3}{4}$
1	10	0.620	0.302	$1\frac{1}{4}$	$1\frac{15}{8}$	$3\frac{1}{2}$	2
	9	0.731	0.420	$1\frac{1}{2}$	$2\frac{1}{2}$	$4\frac{1}{2}$	2
	8	0.837	0.550	1	$2\frac{3}{4}$	$5\frac{1}{2}$	2
$1\frac{1}{8}$	7	0.940	0.694	$1\frac{1}{2}$	$2\frac{3}{4}$	$6\frac{1}{2}$	2
$1\frac{1}{4}$	7	1.065	0.893	2	$2\frac{15}{8}$	$7\frac{1}{2}$	2
$1\frac{3}{8}$	6	1.160	1.057	$2\frac{1}{8}$	$2\frac{17}{8}$	$8\frac{1}{2}$	2
$1\frac{1}{2}$	6	1.284	1.295	$2\frac{3}{8}$	$2\frac{31}{8}$	$9\frac{1}{2}$	2
$1\frac{3}{4}$	$5\frac{1}{2}$	1.389	1.515	$2\frac{9}{8}$	$2\frac{33}{8}$	$10\frac{1}{2}$	2
2	5	1.490	1.744	$2\frac{3}{4}$	$3\frac{3}{8}$	$11\frac{1}{2}$	2
$2\frac{1}{8}$	5	1.615	2.049	$2\frac{11}{8}$	$3\frac{11}{8}$	$12\frac{1}{2}$	2
$2\frac{1}{4}$	$4\frac{1}{2}$	1.712	2.302	$3\frac{1}{8}$	$3\frac{5}{8}$	$13\frac{1}{2}$	2
$2\frac{3}{4}$	$4\frac{1}{2}$	1.962	3.023	$3\frac{1}{2}$	$4\frac{1}{8}$	$14\frac{1}{2}$	2
3	4	2.176	3.719	$3\frac{7}{8}$	$4\frac{1}{2}$	$15\frac{1}{2}$	2
$3\frac{1}{4}$	4	2.426	4.620	$4\frac{1}{4}$	$4\frac{9}{8}$	6	2
$3\frac{1}{2}$	$3\frac{1}{2}$	2.629	5.428	$4\frac{3}{8}$	5	$6\frac{1}{2}$	2
4	$3\frac{1}{2}$	2.879	6.510	5	$5\frac{1}{8}$	$7\frac{1}{2}$	2
$4\frac{1}{2}$	$3\frac{1}{2}$	3.100	7.548	$5\frac{3}{8}$	$6\frac{1}{4}$	$8\frac{1}{2}$	2

to 2" are available. A bill of material should specify the diameter, length, head and nut.

Coupling Bolts.—Coupling bolts differ from machine bolts in that the bodies of the bolts are milled to exactly fit reamed holes of the same nominal diameter as the bolts, and the heads and nuts are faced perpendicular to the axis of the bolt. Stock forms have hexagonal heads and nuts only. The diameters run from $\frac{1}{2}$ " up to $1\frac{1}{4}$ ", varying by $\frac{1}{8}$ ", and lengths from 2" up to 6", varying by $\frac{1}{4}$ ". Specify diameter and length in the bill of material.

Tap Bolts.—Tap bolts, Fig. 6(b), are sometimes used instead of machine bolts where the work is of such a nature that the bolt cannot pass through to receive the nut. The threaded end of the bolt enters a portion of the work itself which serves as a nut. Wherever the work requires frequent disconnecting a stud is to be preferred. Tap bolts are carried in stock with United States Standard thread and standard hexagonal or square heads. Stock diameters run from $\frac{1}{4}$ " to 1" and lengths from $1\frac{1}{2}$ " to 4". Specify diameter, length and head in a bill of material.

Studs.—Studs, Fig. 6(c), serve a similar purpose and in general are to be preferred to tap bolts. This is especially true if the work has to be disconnected frequently. That portion of the work which receives the threaded end of the stud serves as a head while the portion of the work through which the stud passes is held in place by the nut. The stud remains permanently in place, and wear on the threads in a principal member is avoided. The stock form has United States Standard thread and standard hexagonal or square nut. Stock diameters run from $\frac{3}{8}$ " to $1\frac{1}{4}$ " and lengths from $1\frac{1}{2}$ " to 10". Specify the diameter, length and nut in a bill of material.

Stud Bolts.—Stud bolts differ from studs in that both ends are alike and are fitted with a nut on each. They are used instead of machine bolts when projecting portions of the material so cover the bolt hole that the bolt could be inserted only by passing the bolt head through the bolt hole. The second nut acts as a detachable head. Bolts of extreme length are made in this form as it is easier to thread the end of a long rod than to forge a head on it. The stock forms and specifications are the same as for studs.

Automobile Bolts.—The foregoing forms of bolts do not fulfill the exacting requirements of strength and space limits demanded in automobile work. The ever increasing use of automobiles and the necessity for interchangeability in bolts and screws likely to be broken or lost has led to the adoption of a special standard for such parts by the Society of Automobile Engineers. Their standard bolts have hexagonal heads and nuts and United States Standard thread but differ from the usual form in several points, such as finer threads, smaller heads and nuts, heads slotted for screw-driver, end

of bolts drilled and nuts recessed to lock with cotter pins, and shoulder on face of heads and nuts to prevent contact of corners while being screwed up. The body of the bolt is finished to $D-0.001''$, and $S = 1\frac{1}{2} D$. These forms are illustrated in Fig. 7 and the numerical values of the dimensions for the stock sizes are given in Table II.

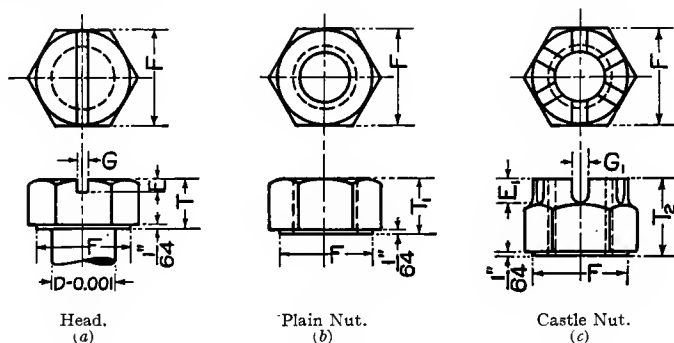


FIG. 7.

TABLE II.
S. A. E. STANDARD SCREWS AND NUTS.

Diameter of Bolt.	Threads per Inch.	Distance across Flats.	Thickness			Depth of Slots		Width of Slots		Diameter of Cotter Pin.
			of Head.	of Nut (plain).	of Nut (castle).	in Head.	in Nut (castle).	in Head.	in Nut (castle).	
D	T.P.I.	F	T	T ₁	T ₂	E	E ₁	G	G ₁	
$\frac{1}{4}$	28	$\frac{7}{16}$	$\frac{3}{16}$	$\frac{7}{32}$	$\frac{9}{32}$	$\frac{3}{32}$	$\frac{3}{32}$	$\frac{1}{16}$	$\frac{5}{64}$	$\frac{1}{16}$
$\frac{5}{16}$	24	$\frac{9}{16}$	$\frac{15}{64}$	$\frac{17}{64}$	$\frac{21}{64}$	$\frac{7}{64}$	$\frac{3}{32}$	$\frac{1}{16}$	$\frac{5}{64}$	$\frac{1}{16}$
$\frac{3}{8}$	20	$\frac{5}{8}$	$\frac{21}{64}$	$\frac{25}{64}$	$\frac{31}{64}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{3}{32}$
$\frac{1}{2}$	20	$\frac{3}{4}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{9}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{3}{32}$
$\frac{9}{16}$	18	$\frac{7}{8}$	$\frac{27}{64}$	$\frac{31}{64}$	$\frac{39}{64}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{5}{32}$	$\frac{1}{8}$
$\frac{5}{8}$	18	$\frac{15}{16}$	$\frac{15}{32}$	$\frac{35}{64}$	$\frac{23}{32}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{32}$	$\frac{5}{32}$	$\frac{1}{8}$
$\frac{11}{16}$	16	1	$\frac{33}{64}$	$\frac{19}{16}$	$\frac{49}{64}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{32}$	$\frac{5}{32}$	$\frac{1}{8}$
$\frac{3}{4}$	16	$1\frac{1}{16}$	$\frac{63}{64}$	$\frac{39}{32}$	$\frac{64}{64}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{32}$	$\frac{5}{32}$	$\frac{1}{8}$
$\frac{7}{8}$	14	$1\frac{1}{4}$	$\frac{127}{128}$	$\frac{79}{64}$	$\frac{127}{128}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{32}$	$\frac{5}{32}$	$\frac{1}{8}$
1	14	$1\frac{7}{8}$	$\frac{255}{128}$	$\frac{159}{64}$	$\frac{255}{128}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{32}$	$\frac{5}{32}$	$\frac{1}{8}$
$1\frac{1}{8}$	12	$1\frac{5}{8}$	$\frac{511}{256}$	$\frac{319}{128}$	$\frac{511}{256}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{32}$	$\frac{5}{32}$	$\frac{1}{8}$
$1\frac{1}{4}$	12	$1\frac{3}{4}$	$\frac{1023}{512}$	$\frac{639}{256}$	$\frac{1023}{512}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{32}$	$\frac{5}{32}$	$\frac{1}{8}$
$1\frac{3}{8}$	12	2	$\frac{2047}{1024}$	$\frac{1279}{512}$	$\frac{2047}{1024}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{32}$	$\frac{5}{32}$	$\frac{1}{8}$
$1\frac{1}{2}$	12	$2\frac{1}{2}$	$\frac{4095}{2048}$	$\frac{2559}{1024}$	$\frac{4095}{2048}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{3}{32}$	$\frac{5}{32}$	$\frac{1}{8}$

8. Nut Locks.—Wherever bolts on structures or machinery are subjected to jarring or vibration there is a tendency for the nuts to work loose and come off. This is especially true where the nature of the work does not permit the nut to be set up hard to get the full advantage of frictional resistance. In such cases it is necessary to provide some form of nut lock. There are a large number of different forms of lock nuts in use. These vary widely in complexity and costliness. Three only of the more common forms will be discussed here.

Jam Nuts.—Probably the commonest form of nut lock is that shown in Fig. 8(a). A second nut is placed on the bolt and screwed up hard against the first to prevent it from turning. It is not necessary that both of these nuts shall be of standard thickness

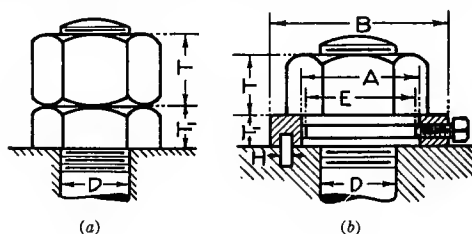


FIG. 8.

and some controversy has arisen as to which should be reduced. A study of the theory in each case will reveal evidence in support of both methods but in general the load is carried by the outer nut when fully set up. If $T = D$ then $T_1 = \frac{1}{2} D$ to D is permissible.

Castellated Nuts.—The outer end of this form of nut is prolonged and through this cylindrical prolongation diametrical slots are milled as illustrated in Fig. 7(c). A hole drilled diametrically through the end of the bolt permits a split cotter pin to be passed through the bolt with its ends projecting into the slot on either side, thus providing a positive lock to the nut. The advantage of the positive locking is overcome for some purposes by the necessity of turning through 60° between the successive locking positions. In the discussion of automobile bolts in the preceding article mention was made of a provision for locking the nut by means of a cotter pin. This is perhaps the most common application of

the castellated nut. The stock sizes correspond to the sizes of automobile bolts given in Table II on page 16 and they are obtainable either soft or casehardened.

Marine Nut Lock.—The marine nut lock requires a special form of nut with a cylindrical projection on the under side extending through a separate locking ring, as shown in Fig. 8(b). The locking ring is prevented from turning by a pin set in the adjacent member and a set screw in the ring is clamped against the projection on the nut to secure it. A groove is cut in this projection to prevent any burr caused by the cup point of the set screw from interfering with the free rotation of the nut after the screw is loosened. When the bolt is placed near the outside of the work as in connecting rods the lock ring is omitted, the bolt hole being counter-bored to receive the projection on the under side of the nut, and a threaded hole for the set screw being tapped through from the outside.

The following empirical equations give satisfactory proportions where D is the nominal diameter of the bolt and G the diameter of the set screw.

$$A = 1\frac{1}{2} D - \frac{1}{16}''.$$

$$B = A + 4 H.$$

$$E = A - \frac{1}{8}''.$$

$$G = \frac{1}{8} D + \frac{1}{8}''.$$

$$H = \frac{1}{8} D + \frac{1}{16}''.$$

$$T = \frac{3}{4} D \text{ to } D.$$

$$T_1 = 2 G.$$

9. Washers.—A washer should be placed underneath a nut or bolt head when by reason of the nature of the material or the size of the bolt hole a suitable bearing surface is not otherwise provided. When the bearing is on metal, a washer cut or punched from wrought iron or steel plate is used. The standard proportions adopted by the manufacturers on October 9, 1895 are given in Table III.

When the bearing is on wood or masonry it is desirable, unless the load be light, to distribute the pressure over a greater area and a cast iron washer of a form shown in Fig. 9 should be used. Good proportions for these washers are given in Table IV. The nomi-

nal diameter of a washer is the same as the diameter of the bolt with which it is used. Specify the nominal diameter of the washer.

TABLE III.
STANDARD WROUGHT IRON AND STEEL CUT WASHERS.

Nominal Diameter of Washer.	Diameter of Hole.	Outside Diameter of Washer.	Thickness (wire gage).	Thickness (approx.).	Nominal Diameter of Washer.	Diameter of Hole.	Outside Diameter of Washer.	Thickness (wire gage).	Thickness (approx.).
$\frac{3}{16}$	$\frac{1}{8}$	$\frac{9}{16}$	18	$\frac{3}{64}$	1	$1\frac{1}{16}$	$2\frac{1}{2}$	9	$\frac{5}{32}$
$\frac{1}{4}$	$\frac{1}{8}$	$\frac{1}{2}$	16	$\frac{1}{16}$	$1\frac{1}{2}$	$1\frac{1}{8}$	$2\frac{3}{4}$	9	$\frac{3}{8}$
$\frac{5}{16}$	$\frac{1}{8}$	$\frac{1}{2}$	16	$\frac{1}{16}$	$1\frac{3}{4}$	$1\frac{3}{8}$	3	9	$\frac{3}{8}$
$\frac{3}{8}$	$\frac{1}{4}$	1	14	$\frac{1}{8}$	2	$1\frac{1}{2}$	$3\frac{1}{4}$	8	$\frac{3}{4}$
$\frac{7}{16}$	$\frac{1}{4}$	$1\frac{1}{4}$	14	$\frac{5}{64}$	$2\frac{1}{2}$	$1\frac{3}{4}$	$3\frac{1}{2}$	8	$\frac{1}{2}$
$\frac{1}{2}$	$\frac{1}{2}$	$1\frac{3}{4}$	12	$\frac{3}{32}$	3	$2\frac{1}{8}$	$3\frac{3}{4}$	8	$\frac{11}{16}$
$\frac{5}{8}$	$\frac{3}{8}$	$1\frac{1}{2}$	12	$\frac{3}{32}$	$3\frac{1}{2}$	$2\frac{1}{4}$	4	8	$\frac{1}{2}$
$\frac{3}{4}$	$\frac{1}{2}$	$1\frac{3}{4}$	10	$\frac{1}{8}$	4	$2\frac{3}{8}$	$4\frac{1}{4}$	8	$\frac{11}{16}$
$\frac{7}{8}$	$\frac{3}{4}$	2	10	$\frac{1}{8}$	$4\frac{1}{2}$	$2\frac{7}{8}$	$4\frac{3}{4}$	8	$\frac{1}{2}$
1	$\frac{7}{8}$	$2\frac{1}{4}$	9	$\frac{3}{16}$	5	$3\frac{1}{2}$	$4\frac{3}{4}$	6	$\frac{9}{16}$

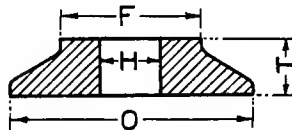


FIG. 9.

TABLE IV.
CAST IRON WASHERS.

Nominal Diameter of Washer.	Diameter of Hole.	Diameter at Base.	Diameter at Top.	Thickness.	Nominal Diameter of Washer.	Diameter of Hole.	Diameter at Base.	Diameter at Top.	Thickness.
D	H	O	F	T	D	H	O	F	T
$\frac{3}{8}$	$\frac{7}{16}$	$2\frac{3}{8}$	$1\frac{1}{2}$	$\frac{9}{16}$	$1\frac{1}{2}$	$1\frac{1}{8}$	$4\frac{3}{4}$	$2\frac{5}{8}$	$1\frac{1}{8}$
$\frac{1}{2}$	$\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{3}{4}$	$\frac{1}{2}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$5\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{1}{4}$
$\frac{5}{8}$	$\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{7}{8}$	$\frac{1}{2}$	1	$1\frac{1}{8}$	$5\frac{3}{4}$	$2\frac{3}{4}$	$1\frac{1}{4}$
$\frac{3}{4}$	$\frac{1}{2}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$\frac{1}{2}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$6\frac{1}{4}$	$3\frac{1}{4}$	$1\frac{1}{2}$
1	$\frac{3}{4}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$\frac{1}{2}$	1	$1\frac{1}{2}$	$7\frac{1}{4}$	$3\frac{1}{2}$	$1\frac{3}{4}$
$1\frac{1}{8}$	$\frac{3}{4}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{1}{2}$	$7\frac{1}{4}$	$3\frac{1}{2}$	$1\frac{3}{4}$
$1\frac{1}{4}$	$\frac{3}{4}$	$2\frac{1}{2}$	$2\frac{1}{2}$	$\frac{1}{2}$	2	$2\frac{1}{8}$	$8\frac{1}{4}$	$4\frac{1}{4}$	2

10. Screws for Metal.—In general the purpose of a screw is to draw together two or more pieces of material by means of its head and the action of its thread when entering the tapped hole in the

last of these pieces. The screw passes freely through holes drilled in each of the other pieces. A screw is not well suited to resist a tendency of the pieces to slide over each other but is sometimes used for that purpose when the load is small. The following symbols will be used in the description of screws.

D = diameter of screw.

L = length from under side of head to extreme end of thread, except for countersunk and French heads and set screws.

S = length of perfect screw thread.

The proportions given here are approximate and for drafting only.

Cap Screws.—Cap screws differ from tap bolts chiefly in their smaller and more finished heads, in the greater number of forms for the heads, and in the relative lengths of their threaded portions. They are to be preferred to them in places where finish is a deciding element. They resemble machine screws in the forms of their

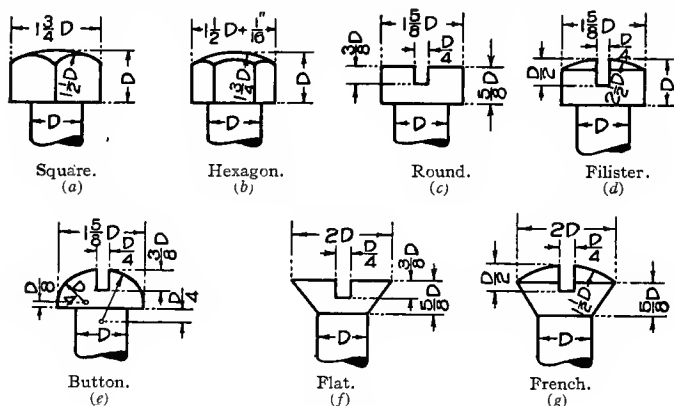


FIG. 10.

heads but the latter are designed for much lighter service. The range of sizes of the two kinds overlap to some extent but owing to the different system of measuring sizes they are in no wise interchangeable.

The proportions for the heads of cap screws have never been standardized and vary slightly as produced by different makers. The proportions given in Fig. 10 are accurate enough for general purposes but wherever exact values are needed they should be obtained from the catalogs of the maker from whom the screws

are to be purchased. These heads differ both in name and in proportions from the similar forms for machine screws adopted by the American Society of Mechanical Engineers.

Stock forms of cap screws have United States Standard or full V thread, with square, hexagonal, round, filister, button, flat, or French heads, as shown in Fig. 10(a), (b), (c), (d), (e), (f) and (g) respectively. $S = \frac{3}{4} L$ up to 1" diameter by 4" long, and $\frac{1}{2} L$ for larger sizes. Lengths vary by $\frac{1}{4}$ " from $\frac{3}{4}$ " to 5", except that screws with square or hexagonal heads, Fig. 10(a) and (b), are carried in a $\frac{7}{8}$ " length. Diameters and threads per inch are the same as for bolts having the same form of thread. Specify the diameter, form of thread, length and head.

Machine Screws.—The term "machine screw" has been applied to cover numerous forms of small screws having heads slotted so that they may be driven with a screw-driver. The diameter of these screws is designated by a number instead of in inches, there being a uniform difference of slightly less than $\frac{1}{64}$ " in the diameters of the successive numbers. The manufacturers differed so much in the number of threads per inch in these screws that the American Society of Mechanical Engineers has established a standard in order that these screws might be made interchangeable, and that standard only will be treated in this book as it is necessary to consult the catalogs of the respective makers for other

TABLE V.
A. S. M. E. STANDARD MACHINE SCREWS.

Number.	Diameter.	Threads per Inch.	Range of Stock Lengths.	Number.	Diameter.	Threads per Inch.	Range of Stock Lengths.
2	0.086	64	$\frac{3}{16} - \frac{1}{8}$	14	0.242	24	$\frac{1}{2} - 2$
3	0.099	56	$\frac{3}{16} - \frac{1}{8}$	16	0.268	22	$\frac{1}{2} - 2\frac{1}{2}$
4	0.112	48	$\frac{3}{16} - \frac{1}{8}$	18	0.294	20	$\frac{1}{2} - 2\frac{1}{2}$
5	0.125	44	$\frac{3}{16} - \frac{1}{8}$	20	0.320	20	$\frac{1}{2} - 2\frac{1}{2}$
6	0.138	40	$\frac{3}{16} - 1$	22	0.346	18	$\frac{1}{2} - 3$
7	0.151	36	$\frac{1}{4} - 1\frac{1}{8}$	24	0.372	16	$\frac{1}{2} - 3$
8	0.164	36	$\frac{1}{4} - 1\frac{1}{8}$	26	0.398	16	$\frac{1}{2} - 3$
9	0.177	32	$\frac{1}{4} - 1\frac{1}{8}$	28	0.424	14	$\frac{1}{2} - 3$
10	0.190	30	$\frac{1}{4} - 1\frac{1}{8}$	30	0.450	14	$\frac{1}{2} - 3$
12	0.216	28	$\frac{1}{4} - 1\frac{1}{8}$				

standards. The numbers most commonly used, with the threads per inch and available lengths are given in Table V. The diameter of the screw must not exceed the diameter given in the table but may be a few thousandths less. The thread is United States Standard.

The stock forms have flat filister, oval filister, round, flat, or French heads, shown in Fig. 11(a), (b), (c), (d) and (e), all except the last of which have been given standard proportions by the American Society of Mechanical Engineers and exact values are obtainable from tables in Vol. 28 of the Transactions. The pro-

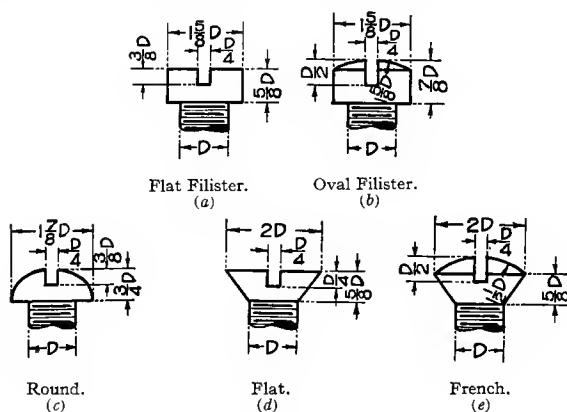


FIG. 11.

portions given in Fig. 11 are sufficiently accurate for most purposes. The heads differ both in name and in proportions from those for cap screws. The stock lengths vary by $\frac{1}{16}$ " from $\frac{3}{16}$ " to $\frac{1}{2}$ ", by $\frac{1}{8}$ " from $\frac{1}{2}$ " to $1\frac{1}{2}$ " and by $\frac{1}{4}$ " from $1\frac{1}{2}$ " to 3". The thread extends the full length of the body of the screw. Specify number, length and head in a bill of material.

Set Screws.—Set screws differ radically from other screws in their action. They pass through a threaded hole in one member until the point of the screw bears firmly against the second member in order to secure a grip to prevent relative sliding between the two members or to push the members apart. The points are made in various shapes, as shown in Fig. 12(a), (b), (c) and (d), and hardened to resist wear. Of these forms the cup point, Fig. 12(a),

is so generally preferred that it is furnished unless otherwise specified. The standard square head which is the form furnished unless

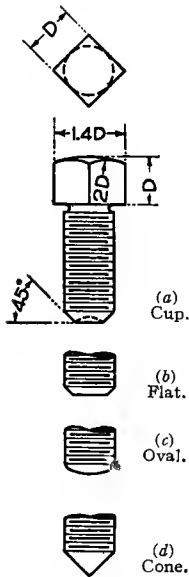


FIG. 12.

otherwise specified, is approximately cubical, as shown in Fig. 12, but low head and headless screws are also furnished if desired. The low head has a thickness of one-half that of the standard square head. Formerly the headless set screws were slotted to be turned up with a screw-driver but in many cases it was impossible to get the screw-driver in position to use and often the screws were split apart in setting up. In a more modern form, sometimes known as a safety set screw, the end is recessed, as shown in Fig. 13, to fit the end of a square, hexagonal or fluted bar of steel which has been bent at right angles to form a wrench by means of which it may be tightened. Stock forms have United States Standard or full V thread with diameter and number of threads per inch the same as for bolts of the same form of thread. The thread extends the full length of the body of the screw. The lengths include point and vary by $\frac{1}{8}$ " from $\frac{1}{2}$ " to 1" and by $\frac{1}{4}$ " from 1" to 5".

Stock diameters run up to $1\frac{1}{4}$ ". Specify the diameter, form of thread, length, point and head in the bill of material.



FIG. 13.

11. Screws for Wood.—Screws used in wood serve the same general purpose as those for metal but they have a distinctly different form of thread and point. The thread resembles the V thread carried down to about one-half depth, leaving the root

flat, as shown in Fig. 14. It is not necessary to know the pitch of the threads as the screw forms the thread in the wood in the process of driving. These screws may have either gimlet point, Fig. 14(a), for driving with a screw-driver, or cone point, Fig. 14(b), for driving with a hammer. The former is standard for most work and is usually supplied unless otherwise specified.

Lag Screws.—Lag screw heads are of the same general form and proportions as the heads for bolts as shown in Fig. 6(a) and (b) on page 13, and the diameters and lengths vary in the same manner up to 1" in diameter and 12" in length. The length is measured from the point to the under side of the head. The stock forms have square or hexagonal heads and gimlet or cone points. They are used in wood when the conditions are similar to those under which tap bolts would be used in metal. Specify the diameter, length, head and point.

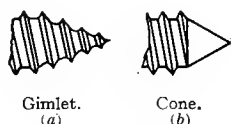


FIG. 14.

Hanger Screws.—Hanger screws differ from lag screws in that the heads have been replaced by nuts the forms and threads of which are the same as shown for bolts in Fig. 6(a) and (c) on page 13. They are used in wood when the conditions are similar to those under which a stud would be used in metal. The stock sizes are the same as for lag screws, but the length is measured from point to the outer end of thread for the nut. The stock forms have one square or hexagonal nut and gimlet or cone point. Specify diameter, length, nut and point.

Wood Screws.—Wood screws are used for light work corresponding to that for which machine screws are used in metal. Their diameters and lengths are measured in the same manner as for machine screws. They are carried in stock with round, flat and French heads and gimlet point only. These forms of heads are the same as those shown for machine screws in Fig. 11(c), (d) and (e) on page 22. The stock diameters and lengths are given in Table V on page 21. Specify number, length and head.

CHAPTER III.

KEYS AND TAPER PINS.

12. **Use of Keys.**—Keys are used primarily to prevent rotation of gears, pulleys, etc., relative to the shafts upon which they are mounted. This is accomplished by fitting the key so that it stands parallel to the axis of the shaft, partly within the shaft and partly within the hub of the gear or pulley as shown in end view in Fig. 15. The grooves or recesses cut in the hub and shaft to receive the key are called **keyways** or **key seats**. Well fitted keys may also prevent in some degree the tendency of the keyed members to slide along the shaft. In heavy machinery subject to shock

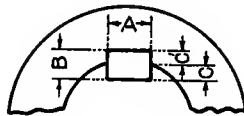


FIG. 15.

two keys placed 90° apart are often used. When so placed they insure the advantage of contact at three places even though the hub and shaft are not accurately fitted.

13. **Forms and Proportions for Keys.**—Special conditions may justify the use of a great variety of forms of keys. Only those forms, however, which have become standardized and are found in general practice will be discussed in this book.

All of these standard forms are of uniform width throughout their lengths. They should fit the sides of the key seats accurately. **Straight keys** are also of uniform thickness while **taper keys** vary uniformly in thickness from end to end. Each of these forms may be either square or rectangular in cross section and are called **square keys** and **flat keys** respectively. The length of the key depends on the length of hub of the keyed-on member and is usually about $\frac{1}{2}$ " longer. When the point of a driven key is not accessible, a gib or head, as shown in Fig. 16, is formed on the other end for the purpose of withdrawing the key, the proportions of the key

remaining otherwise unchanged. The gib is seldom required on a straight key. The measurements of keys and key seats, illustrated in Figs. 15 and 16, are made as follows:

- A = width of key = length of gib,
- B = thickness of key = height of gib,
- C = depth of key seat in shaft,
- C' = depth of key seat in hub,
- L = length of key.

Straight Keys.—Keys without taper are used in machine tools and where accurate centering is required. Theoretically, they

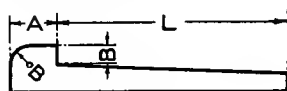


FIG. 16.

should bear on the sides only. Such keys are usually square in cross section. Since the shafts of machine tools are designed for extra stiffness these keys are smaller in proportion to the size of the shaft than are the keys on line shafts which are usually of this same form. This is best shown by a comparison of the sizes given in Tables VI and VII.

TABLE VI.
SQUARE KEYS FOR MACHINE TOOLS.

Diameter of Shaft.	Size of Key.	Diameter of Shaft.	Size of Key.
$-\frac{1}{16}$	$\frac{1}{8} \times \frac{1}{8}$	$2\frac{3}{4} - 3\frac{1}{16}$	$\frac{11}{16} \times \frac{11}{16}$
$1 - 1\frac{3}{16}$	$\frac{3}{16} \times \frac{3}{16}$	$4 - 5\frac{7}{16}$	$\frac{13}{16} \times \frac{13}{16}$
$1\frac{1}{4} - 1\frac{7}{16}$	$\frac{1}{4} \times \frac{1}{4}$	$5\frac{1}{2} - 6\frac{1}{16}$	$\frac{15}{16} \times \frac{15}{16}$
$1\frac{1}{2} - 1\frac{11}{16}$	$\frac{5}{16} \times \frac{5}{16}$	$7 - 8\frac{1}{16}$	$1\frac{1}{16} \times 1\frac{1}{16}$
$1\frac{3}{4} - 2\frac{3}{16}$	$\frac{7}{16} \times \frac{7}{16}$	$9 - 10\frac{1}{16}$	$1\frac{3}{16} \times 1\frac{3}{16}$
$2\frac{1}{4} - 2\frac{1}{16}$	$\frac{9}{16} \times \frac{9}{16}$	$11 - 12\frac{1}{16}$	$1\frac{5}{16} \times 1\frac{5}{16}$

The keyed piece may be prevented from sliding by means of set screws through the hub, bearing on the top of the key, but this method tends to throw the keyed member off center.

Flat keys without taper (except at the point, which may be relieved slightly to assist in starting the key) are fitted to drive, bearing on all four sides. They are used on work subject to shock

TABLE VII.
KEYS FOR SHAFTING.

Diameter of Shaft.	Size of Key.	Diameter of Shaft.	Size of Key.
$\frac{3}{4}$ - $1\frac{1}{4}$	$\frac{1}{4}$ x $\frac{1}{4}$	$5\frac{5}{16}$ - $5\frac{3}{4}$	$1\frac{3}{8}$ x $1\frac{3}{8}$
$1\frac{1}{16}$ - $1\frac{3}{4}$	$\frac{5}{16}$ x $\frac{3}{8}$	$5\frac{13}{16}$ - $6\frac{1}{4}$	$1\frac{1}{2}$ x $1\frac{1}{2}$
$1\frac{13}{16}$ - $2\frac{1}{4}$	$\frac{3}{8}$ x $\frac{1}{2}$	$6\frac{1}{8}$ - $7\frac{1}{4}$	$1\frac{1}{2}$ x $1\frac{1}{2}$
$2\frac{5}{16}$ - $2\frac{3}{4}$	$\frac{3}{8}$ x $\frac{5}{8}$	$7\frac{1}{8}$ - $8\frac{1}{4}$	2 x $1\frac{1}{2}$
$2\frac{13}{16}$ - $3\frac{1}{4}$	$\frac{3}{4}$ x $\frac{3}{4}$	$8\frac{5}{16}$ - $9\frac{1}{4}$	$2\frac{1}{4}$ x $1\frac{1}{2}$
$3\frac{5}{16}$ - $3\frac{3}{4}$	$\frac{7}{8}$ x $\frac{7}{8}$	$9\frac{5}{16}$ - $10\frac{1}{4}$	$2\frac{1}{2}$ x $1\frac{1}{2}$
$3\frac{13}{16}$ - $4\frac{1}{4}$	1 x 1	$10\frac{9}{16}$ - $11\frac{1}{4}$	$2\frac{3}{4}$ x $1\frac{1}{2}$
$4\frac{5}{16}$ - $4\frac{3}{4}$	$1\frac{1}{8}$ x $1\frac{1}{8}$	$11\frac{1}{16}$ - $12\frac{1}{4}$	3 x 2
$4\frac{13}{16}$ - $5\frac{1}{4}$	$1\frac{1}{4}$ x $1\frac{1}{4}$	$12\frac{5}{16}$ - $13\frac{1}{4}$	$3\frac{1}{4}$ x 2

and heavy loads, such as in keying cranks and flywheels to engine shafts where the tendency to push the shaft out of center is resisted by the close fitting of the hub. The thickness is usually about five-eighths of the width. Standard proportions are given in Table VIII.

TABLE VIII.
FLAT KEYS.

Diameter of Shaft.	Size of Key.	Diameter of Shaft.	Size of Key.
-1	$\frac{1}{4}$ x $\frac{5}{8}$	$3\frac{1}{16}$ - $3\frac{1}{2}$	$\frac{7}{8}$ x $\frac{1}{2}$
$1\frac{1}{16}$ - $1\frac{1}{4}$	$\frac{5}{16}$ x $\frac{9}{16}$	$3\frac{9}{16}$ - 4	1 x $\frac{5}{8}$
$1\frac{5}{16}$ - $1\frac{1}{2}$	$\frac{3}{8}$ x $\frac{1}{4}$	$4\frac{1}{16}$ - 5	$1\frac{1}{8}$ x $\frac{11}{16}$
$1\frac{9}{16}$ - $1\frac{3}{4}$	$\frac{7}{16}$ x $\frac{9}{16}$	$5\frac{1}{16}$ - 6	$1\frac{3}{8}$ x $\frac{11}{16}$
$1\frac{13}{16}$ - 2	$\frac{1}{2}$ x $\frac{5}{8}$	$6\frac{1}{16}$ - 7	$1\frac{1}{2}$ x $\frac{7}{8}$
$2\frac{1}{16}$ - $2\frac{1}{2}$	$\frac{5}{8}$ x $\frac{3}{4}$	$7\frac{1}{16}$ - 8	$1\frac{3}{4}$ x 1
$2\frac{9}{16}$ - 3	$\frac{3}{4}$ x $\frac{7}{8}$		

Straight keys, either square or flat, are usually purchased in long bars which have been cold drawn to the exact section desired and from which keys may be cut to any desired length. Specify width, thickness and length in the order named.

Taper Keys.—Taper keys are used to prevent sliding as well as turning of the keyed-on member. They are also used in work subject to shock and heavy loads, such as in keying cranks and flywheels to engine shafts, and for fastening pulleys and gears to shafts, etc. The standard taper for the thickness is $\frac{1}{8}$ " per foot of length. As these keys bear on all sides one of the key seats must also be tapered. This taper is made in the key seat in the hub. Members to be fastened with taper keys should fit the shafts very

closely; otherwise driving the key will throw them off center. The thickness of taper keys is measured at the large end. The form of the cross section at that end may be either square or flat. The proportions given in Tables VII and VIII apply equally well to taper keys.

Tapered square keys, with or without gib, are listed by manufacturers, with cross sections from $\frac{1}{8}$ " to 3" square varying by $\frac{1}{16}$ " and with lengths from $1\frac{1}{2}$ " to 24" varying by $\frac{1}{2}$ ". Flat tapered keys are not listed except as made to order. Specify width, thickness and length in the order named.

For the purpose of comparison, the proportions adopted for use by a few of the larger users of keys are given in Table IX in which D denotes the diameter of the shaft.

TABLE IX.
PROPORTIONS OF KEYS.

	U. S. Navy St'd.	Jones & Laughlin.	Porter-Allen Co.
Width.	$\frac{3}{16}D + \frac{1}{8}"$	$\frac{1}{4}D$	$\frac{1}{4}D + \frac{1}{8}"$
Thickness.	$\frac{3}{32}D + \frac{1}{8}"$	$\frac{5}{32}D + \frac{1}{16}"$	$\frac{1}{11}D + 0.16"$

Feather Keys or Splines.—Where the keyed piece is to slide along the shaft the key is either made long enough to provide for this motion, or it is fastened to the sliding piece and the keyway in the shaft is made long enough to permit the desired amount of sliding. Such keys, called **feather keys** or **splines**, are frequently made thicker than their width as shown by the dimensions given in Table X. This additional thickness is to provide sufficient

TABLE X.
DIMENSIONS OF FEATHER KEYS.

Diameter of Shaft.	Size of Feather.	Diameter of Shaft.	Size of Feather.
—1	$\frac{1}{4}$ x $\frac{3}{8}$	$3\frac{9}{16}$ — 4	1 x $1\frac{1}{4}$
$1\frac{1}{16}$ — $1\frac{1}{4}$	$\frac{1}{8}$ x $\frac{1}{16}$	$4\frac{1}{16}$ — 5	$1\frac{1}{8}$ x $1\frac{1}{8}$
$1\frac{5}{16}$ — $1\frac{1}{2}$	$\frac{3}{8}$ x $\frac{1}{2}$	$5\frac{1}{16}$ — 6	$1\frac{3}{8}$ x $1\frac{3}{8}$
$1\frac{9}{16}$ — $1\frac{3}{4}$	$\frac{7}{8}$ x $\frac{9}{16}$	$6\frac{1}{16}$ — 7	$1\frac{1}{2}$ x $1\frac{3}{4}$
$1\frac{13}{16}$ — 2	$\frac{1}{2}$ x $\frac{5}{8}$	$7\frac{1}{16}$ — 8	$1\frac{3}{4}$ x 2
$2\frac{1}{16}$ — $2\frac{1}{2}$	$\frac{3}{4}$ x $\frac{3}{4}$	$8\frac{1}{16}$ — 9	2 x $2\frac{1}{2}$
$2\frac{5}{16}$ — 3	$\frac{3}{4}$ x $\frac{7}{8}$	$9\frac{1}{16}$ — 10	$2\frac{1}{4}$ x $2\frac{3}{4}$
$3\frac{1}{16}$ — $3\frac{1}{2}$	$\frac{7}{8}$ x 1		

surface to withstand the wear due to the sliding contact and to resist the more severe load conditions due to the looser fitting of the feather key. Two keys, placed on opposite sides of the shaft, are sometimes used to improve the conditions for sliding. They are fastened in the shaft by sunk round head cap screws, the diameter of the heads being about three-fourths the width of the feather. The screws should enter the shaft a distance equal to the diameter of the screw. Where a feather key is placed at the end of a shaft it is sometimes dovetailed into the shaft.

Another form of feather key is designed to fit accurately into a keyway of the form shown in Fig. 19(b) on page 34. They are set into the shaft a depth equal to the width of the key and are fitted to drive. No additional fastening is necessary to secure them in place. These feather keys may be purchased in standard sizes designated by numbers. These numbers are the same as those given in Table XI for Woodruff Standard Keys and the widths and lengths agree with the values there given for the widths and diameters for the corresponding numbers. Specify the number of the key.

14. Woodruff System of Keys.—The Woodruff keys consist of circular segments, as shown in Fig. 17, which are set in keyways cut

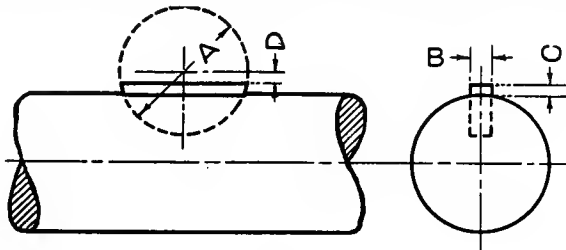


FIG. 17.

in the shafts by means of milling cutters of the same radius as the segment. They have the advantage of adjusting themselves perfectly to any taper of the keyway in the hub, but there is also the disadvantage that the keyed-on member must always be forced on over the key after the key is in position. These keys are made in a large number of standard sizes, designated by numbers and letters. Fig. 17 shows the form of the shorter keys, the proportions

for which are given in Table XI. The sizes suitable for use in the various diameters of shafts are given in Table XII.

TABLE XI.
PROPORTIONS FOR WOODRUFF STANDARD KEYS.

Number of Key.	Diameter of Key.	Width of Key.	Depth of Keyway.	Distance from Top to Center of Key.	Number of Key.	Diameter of Key.	Width of Key.	Depth of Keyway.	Distance from Top to Center of Key.
	A	B	C	D		A	B	C	D
1	$\frac{1}{2}$	$\frac{1}{16}$	$\frac{1}{32}$	$\frac{3}{64}$	B	1	$\frac{5}{16}$	$\frac{5}{32}$	$\frac{1}{16}$
2	$\frac{1}{2}$	$\frac{3}{32}$	$\frac{3}{64}$	$\frac{3}{64}$	16	$1\frac{1}{8}$	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{5}{64}$
3	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{1}{16}$	$\frac{3}{64}$	17	1	$\frac{7}{32}$	$\frac{7}{64}$	$\frac{5}{64}$
4	$\frac{1}{2}$	$\frac{3}{16}$	$\frac{1}{8}$	$\frac{1}{16}$	18	1	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{1}{16}$
5	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{1}{16}$	$\frac{1}{16}$	C	$1\frac{1}{8}$	$\frac{5}{16}$	$\frac{5}{32}$	$\frac{1}{16}$
6	$\frac{1}{2}$	$\frac{5}{32}$	$\frac{5}{64}$	$\frac{1}{16}$	19	$1\frac{1}{4}$	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{5}{64}$
7	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{1}{16}$	$\frac{1}{16}$	20	$1\frac{1}{4}$	$\frac{7}{32}$	$\frac{7}{64}$	$\frac{5}{64}$
8	$\frac{1}{2}$	$\frac{3}{16}$	$\frac{3}{64}$	$\frac{1}{16}$	21	$1\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{5}{64}$
9	$\frac{1}{2}$	$\frac{1}{8}$	$\frac{1}{16}$	$\frac{1}{16}$	D	$1\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{32}$	$\frac{5}{64}$
10	$\frac{1}{2}$	$\frac{5}{32}$	$\frac{5}{64}$	$\frac{1}{16}$	E	$1\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{7}{64}$
11	$\frac{7}{8}$	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{1}{16}$	22	$1\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{3}{32}$
12	$\frac{7}{8}$	$\frac{7}{32}$	$\frac{7}{64}$	$\frac{1}{16}$	23	$1\frac{3}{8}$	$\frac{5}{16}$	$\frac{5}{32}$	$\frac{3}{32}$
A	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{1}{16}$	F	$1\frac{5}{8}$	$\frac{5}{8}$	$\frac{1}{16}$	$\frac{3}{32}$
13	1	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{1}{16}$	24	$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{5}{64}$
14	1	$\frac{7}{16}$	$\frac{7}{32}$	$\frac{1}{16}$	25	$1\frac{1}{2}$	$\frac{5}{16}$	$\frac{5}{32}$	$\frac{7}{64}$
15	1	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{1}{16}$	G	$1\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$	$\frac{7}{64}$

TABLE XII.

Diameter of Shaft.	Number of Key.	Diameter of Shaft.	Number of Key.
$\frac{5}{16} - \frac{3}{8}$	1	$1\frac{3}{16}$	11, 13, 16
$\frac{7}{16} - \frac{1}{2}$	2, 4	$1\frac{1}{4} - 1\frac{5}{16}$	12, 14, 17, 20
$\frac{9}{16} - \frac{5}{8}$	3, 5	$1\frac{3}{8} - 1\frac{1}{2}$	14, 17, 20
$\frac{11}{16} - \frac{3}{4}$	3, 5, 7	$1\frac{1}{2} - 1\frac{5}{8}$	15, 18, 21, 24
$\frac{13}{16} - \frac{7}{8}$	6, 8	$1\frac{11}{16} - 1\frac{3}{4}$	18, 21, 24
$\frac{1}{8} - 1\frac{5}{16}$	6, 8, 10	$1\frac{13}{16} - 2$	23, 25
1	9, 11, 13	$2\frac{1}{16} - 2\frac{1}{2}$	25
$1\frac{1}{16} - 1\frac{1}{8}$	9, 11, 13, 16		

In the longer keys the distance, D, is considerably increased, otherwise the form of the key differs only in the squaring of the ends that project above the shaft. Sometimes, where longer keys

are desirable, two or more keys of the shorter form are set in the shaft end to end. Specify the number of key in the bill of material.

15. Keyways or Key Seats.—Practice is not wholly uniform as to the relative depths of the keyways in the shaft and in the hub. In the United States the ordinary practice is to make the depth of each equal to one-half the thickness of the key. This depth is measured at the side of the keyways as shown in Fig. 15 on page 25. For a taper key the keyway in the hub is tapered and its depth is measured at the deeper end.

16. Taper Pins.—Taper pins are used to prevent rotational or axial sliding between bodies through which they pass. The holes for tapered pins are reamed to the same taper as the pin and the pin is driven tight. They may also be used in the place of keys in light work. Standard taper pins (usually called "Pratt & Whitney Standard Taper Pins") have a uniform taper of $\frac{1}{4}$ " in diameter per

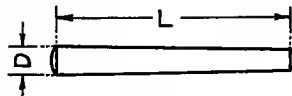


FIG. 18.

foot of length. The length is measured from end to end of the uniform taper as shown in Fig. 18, the ends being convex. They are made in standard diameters designated by numbers and with

TABLE XIII.
PROPORTIONS FOR TAPER PINS.

Number.	Length, L.		Diameter, D.		Number.	Length, L.		Diameter, D.	
	Minimum.	Maximum.	Exact.	Approx.		Minimum.	Maximum.	Exact.	Approx.
00	$\frac{1}{4}$	$2\frac{1}{4}$	0.136	$\frac{9}{64}$	5	$\frac{3}{4}$	4	0.289	$\frac{19}{64}$
0	$\frac{1}{2}$	3	0.156	$\frac{5}{32}$	6	$\frac{3}{4}$	5	0.341	$\frac{1}{2}$
1	$\frac{3}{4}$	3	0.172	$\frac{11}{64}$	7	1	5	0.409	$\frac{13}{32}$
2	$1\frac{1}{4}$	$3\frac{1}{2}$	0.193	$\frac{3}{16}$	8	$1\frac{1}{4}$	5	0.492	$\frac{1}{2}$
3	$1\frac{3}{4}$	3	0.219	$\frac{7}{32}$	9	$1\frac{3}{4}$	6	0.591	$\frac{19}{32}$
4	2	4	0.250	$\frac{1}{4}$	10	$1\frac{3}{4}$	6	0.706	$\frac{9}{16}$

lengths varying by $\frac{1}{4}$ ". The exact and approximate diameters of the large end and the stock lengths for the different sizes are given in Table XIII.

The limits of the stock lengths and diameters listed by different makers vary slightly. The range included in Table XIII may be considered representative of ordinary practice. Specify number and length of pin.

CHAPTER IV.

SHAFTING AND SHAFT FITTINGS.

17. **Shafting.**—A shaft consists of a long, cylindrical bar of wrought iron or steel so mounted in bearings that it may rotate about its own longitudinal axis, transmitting this rotation to other machine parts which are attached to it. That portion of the shaft which lies within the bearing is designated as the **journal**. The bearings are sometimes called **journal boxes** or simply **boxes**. Shafting is the term applied to the stock material of cylindrical form from which shafts are made. Formerly wrought iron was the material chiefly used. It was rolled when hot into cylindrical bars and when cold turned in lathes to exact size and polished. The stock sizes of the hot rolled bars varied by $\frac{1}{4}$ " and they were reduced $\frac{1}{16}$ " in finishing thus establishing a set of stock sizes for shafting varying by $\frac{1}{4}$ " but always $\frac{1}{16}$ " less than each even $\frac{1}{4}$ " in diameter. In later years the wrought iron shafting has been largely supplanted by steel which has been rolled to exact size when cold, and is known as "cold rolled shafting."

Increased facilities of manufacture and more exact methods of design have led to more stock sizes. These vary by smaller intervals and start from the even inches as a base. In all cases, however, where the interval in these stock sizes is greater than $\frac{1}{16}$ " the old standard sizes are maintained in addition. The lists of some of the representative dealers give diameters varying by $\frac{1}{16}$ " from $\frac{3}{16}$ " up to 4" and by $\frac{1}{8}$ " up to 5". Shafts above 5" in diameter are usually forged to order. The commercial lengths vary greatly but approximate 24 feet for full length pieces, with 30 feet as a maximum. Dealers furnish shafting to specified lengths. A small extra charge is made for pieces under 12" or over 24 feet in length. These limits vary somewhat with the different dealers.

18. **Keyways.**—Machine parts which are attached to a shaft may depend wholly upon the tightness of their grip and friction for the driving power received but, except for light loads, it is customary to make the connection positive by inserting keys as

described in article 15 on page 31. Owing to the improved facilities possessed by the makers and the larger dealers for cutting the keyways in shafting, it is usually advisable to purchase shafting with the keyways cut to order. These keyways may be cut with an ordinary milling cutter having a width equal to the width of the key, which leaves the ends curved to the radius of the cutter as shown in Fig. 19(a), or they may be cut with an "end mill" having a diameter equal to the width of the key. In keyways cut with an end mill the ends may be left semi-circular as shown in Fig. 19(b), or, after cutting with the "end mill," the ends of the keyway may be cut out square as shown in Fig. 19(c), by chipping with a cape

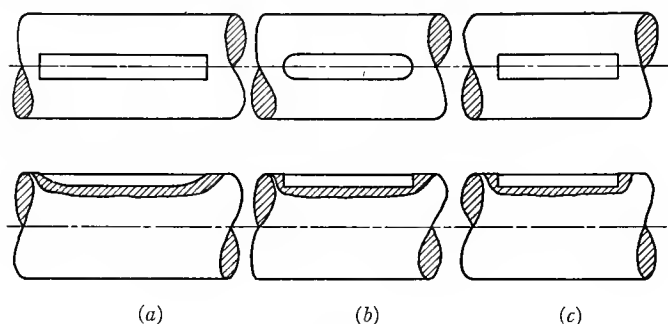


FIG. 19.

chisel. These forms are increasingly expensive in the order given on account of the additional labor and equipment required. Keyways of the second form of limited size may be cheaply made in small shafts for machine tools.

19. Couplings.—As stated in article 17, shafting is made in relatively short pieces which must be fastened together end to end when in place so that their centers shall at all times be truly aligned. Many of these fastenings may be of a permanent nature while others must be of a nature to permit disconnection at frequent intervals. Those of the latter class are commonly called clutches or clutch couplings.

20. Permanent Couplings.—There are several types of permanent couplings. For very light work a **sleeve coupling** may be used. This consists of a simple sleeve, as shown in Fig. 20, into

which the ends of the shafts may be inserted and secured by set screws. It is necessary that the heads of these set screws shall be sunk into the sleeve or that headless set screws be used to avoid



FIG. 20.



FIG. 21.

danger to workmen. Sometimes, where keyways in the shaft are objectionable or have not been provided, **keyless couplings** are used in which the grip on the shaft is frictional only. The necessary pressure is obtained by forcing the halves of a sleeve together by means of rings driven over the tapered ends of the sleeve as shown in Fig. 21, or by forcing a sleeve on over a tapered bushing by means of bolts as illustrated in Fig. 22.

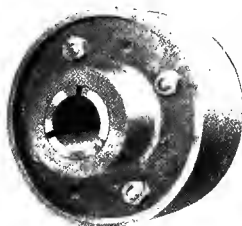


FIG. 22.

Where a more positive connection is desired keys may be used with a sleeve in which case it is made in halves, as shown in Fig. 23, to bolt together over the keys when they are in place. It would

otherwise be necessary to make the keyways in the shafts extend beyond the sleeve an amount equal to the length of the key, to



FIG. 23.

allow the key to be inserted. Any of these couplings which tighten up on the shafts are known as **compression couplings**.



FIG. 24.

A positive connection may also be obtained by the use of **flanged couplings**, illustrated in Fig. 24 and shown in section in Fig. 25. These couplings are generally finished all over and the overhanging

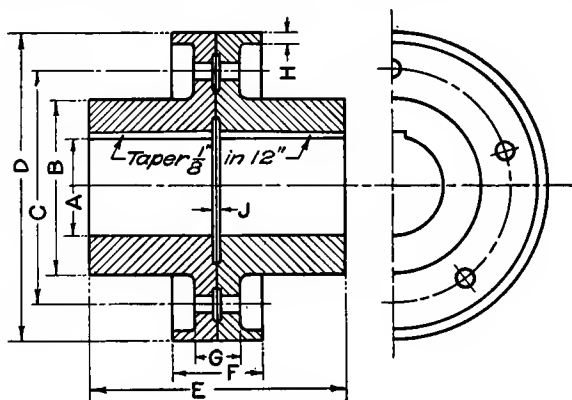


FIG. 25.

rim prevents danger from the clothing of workmen catching on the bolt heads or nuts. The bolt holes are reamed for coupling bolts as described in article 7 on page 14. These bolts are made to fit the holes accurately in order that each bolt shall take its share of the load. The accurate centering of the shafts is secured by extending one of them entirely through its coupling until it enters the mating coupling on the other shaft or, at an added expense, the face of one of the flanges may be recessed to receive a corresponding projection on the face of the other as shown in section in Fig. 26.

The following empirical equations were derived from the commercial cast iron couplings of one of the leading makers. The

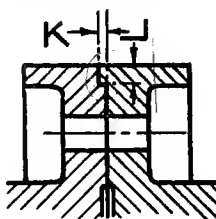


FIG. 26.

dimensions determined from these equations are to be taken to the next larger or the nearest $\frac{1}{16}$ " or $\frac{1}{8}$ " according to the judgment of the draftsman.

A = diameter of shaft.

$$B = 1\frac{2}{3}A + \frac{1}{2}''.$$

$$C = 1\frac{2}{3}A + 2d + 1\frac{1}{8}''.$$

$$D = 1\frac{5}{8}A + 4d + 1\frac{3}{8}''.$$

$$E = 2\frac{3}{16}A + 1\frac{3}{4}''.$$

$$F = l + d.$$

$$G = \frac{1}{4}A + \frac{13}{16}'' \text{ (sizes } 1\frac{1}{4}'' - 3\frac{1}{2}''),$$

$$= \frac{3}{8}A + \frac{3}{8}'' \text{ (sizes } 3\frac{3}{4}'' - 12'').$$

$$H = \frac{1}{16}A + \frac{1}{4}''.$$

$$J = \frac{1}{32}A + \frac{1}{32}'' \text{ but not less than } \frac{1}{8}''.$$

$$K = \frac{1}{32}A + \frac{1}{16}'' \text{ but not less than } \frac{1}{8}''.$$

$$L = \frac{1}{16}A + \frac{1}{4}''.$$

n = number of bolts = $\frac{2}{3}A + 3$.

d = diameter of bolts = $\frac{1}{8}A + \frac{5}{16}$ ".

l = length of bolts = $G + d + \frac{1}{8}$ ".

21. Clutch Couplings.—There are two general classes of clutch couplings; jaw clutches, in which there is a positive connection made by the interlocking of projecting parts, called jaws; and friction clutches, in which the connection is wholly frictional and produced by forcing blocks of wood or other material firmly against a disk of metal.

Jaw Clutches are made either with square jaws, Fig. 27(a), or with spiral jaws, Fig. 27(b). One set of jaws is keyed firmly to the end of one of the shafts while the mating set slides on a feather key in the other shaft to mesh or release as desired. The number of

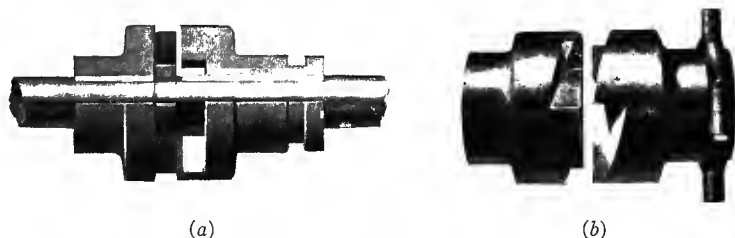


FIG. 27.

jaws is usually two or three, varying with the size of the clutch. The angles between the successive jaws must be accurately constructed in order that the load may be properly divided between the jaws. Those with spiral jaws are more easily thrown in but will drive in one direction only, and hence must be made in right-hand and left-hand shapes. Square jawed clutches will operate in either direction but require a small amount of backlash (rotational play between the jaws when in mesh) to facilitate throwing in. It is necessary that the end of the driven shaft shall be continuously sustained in alignment with the driving shaft. For this purpose there is inserted into the jaw which is fixed to its shaft a finished ring into which the end of the other shaft projects, as shown in Fig. 27(a). This ring is made separately and afterward fastened in place, in order to facilitate the machining of the jaws. Since the ring turns on the end of the shaft when the clutch is thrown out,

oil holes must be provided for lubrication. It is usual to provide an oil hole in each space between the jaws so that one may always be available when oiling without turning over a heavy shaft.

The sectional drawing, Fig. 28, shows a clutch with three square jaws and with the finished ring removed. The proportions given in the following empirical equations were derived from the dimen-

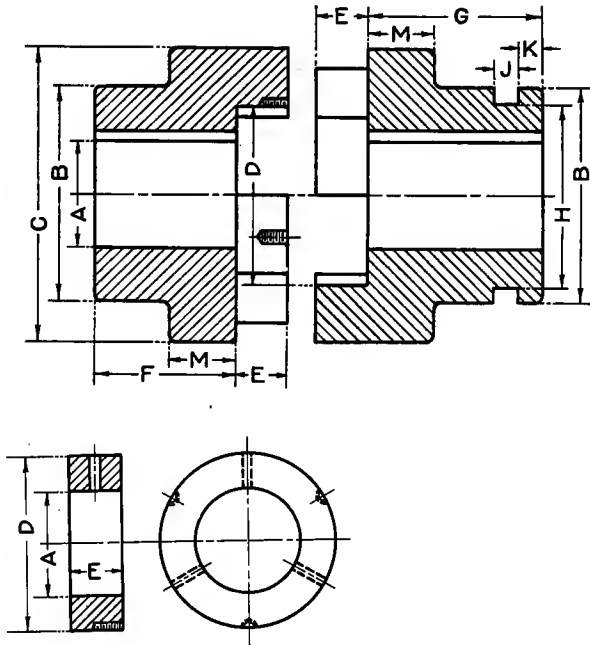


FIG. 28.

sions of commercial cast iron clutches sold by one of the leading makers and are suitable for either square or spiral jawed clutches. A backlash of 2° is allowed in the square jawed clutches.

A = diameter of shaft.

$$B = 1\frac{3}{4}A + \frac{3}{4}''.$$

$$C = 2\frac{3}{8}A + 1\frac{1}{8}''.$$

$$D = 1\frac{1}{2}A + \frac{1}{2}''.$$

$$E = \frac{3}{8}A + \frac{3}{8}''.$$

$$F = A + 1".$$

$$G = 1\frac{1}{8} A + 1\frac{1}{2}".$$

$$H = 1\frac{5}{8} A + \frac{1}{4}".$$

$$J = \frac{1}{10} A + \frac{3}{8}".$$

$$K = \frac{1}{10} A + \frac{1}{4}".$$

$$M = \frac{1}{2} A + \frac{3}{8}".$$

The finished ring is also of cast iron and is held in position by means of headless set screws placed one at each jaw, with half its diameter in the ring and half in the jaw. The points of these screws should



FIG. 29.

bear in order that the screws may be tightened. The following table gives the sizes of set screws used in the different sizes of clutches.

Shaft.	Set Screw.	
Diameter.	Diameter.	Length.
$\frac{15}{16} - 1\frac{15}{16}$	$\frac{1}{4}$	$\frac{1}{2}$
$2\frac{3}{16} - 3\frac{15}{16}$	$\frac{3}{8}$	$\frac{5}{8}$
$4\frac{7}{16} - 6\frac{15}{16}$	$\frac{1}{2}$	$\frac{3}{4}$

The sliding jaw is given an overtravel of $\frac{1}{4}"$ for shafts up to $1\frac{7}{16}"$ diameter, $\frac{3}{8}"$ from $1\frac{1}{2}"$ to $4\frac{1}{16}"$, and $\frac{1}{2}"$ above 5".

Jaw clutches are thrown in or out by means of a lever or shifter of the general form illustrated in Fig. 29. This is connected to a collar, Fig. 27(b), that runs in a groove in the hub of the part of the clutch which slides on the feather key. This collar is of cast iron made in two pieces to bolt together in the groove as is shown in the key drawing, Fig. 30, to which the following empirical proportions apply.

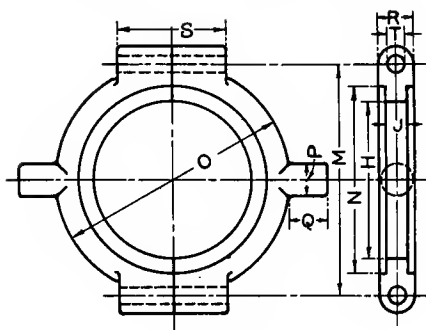


FIG. 30.

A = diameter of shaft.

H = (see Fig. 28).

J = (see Fig. 28).

M = $1\frac{3}{4}A + 2\frac{1}{4}"$.

N = $B + \frac{1}{16}"$ to $B + \frac{1}{8}"$.

O = $2A + 1\frac{1}{2}"$.

P = $\frac{1}{8}A + \frac{5}{8}"$.

Q = $1\frac{1}{4}P$.

R = $2d + \frac{1}{8}"$.

S = $1\frac{1}{4}A$ but not less than $1\frac{1}{2}"$.

T = $d + \frac{1}{32}"$.

d = diameter of bolts = $\frac{1}{16}A + \frac{5}{16}"$.

l = length of bolts = $S + d + \frac{1}{8}"$.

The hub turns continually in the collar when the feather key is in the driving shaft, or in any case, when the clutch is in. An oil hole must, therefore, be provided in one-half of the collar in such a position that it may be up at whatever angle the lever may be given in setting up.

The lever may be vertical, horizontal, or at any intermediate angle as desired but, in the middle of its throw, it should stand approximately at right angles to the shaft. It is made of wrought iron and tapers to a handle at the end, as in Fig. 29. The fork is forged in parts so that they may be placed over the projections on

the collar and then be bolted to the lever as shown in the key drawing, Fig. 31, to which the following proportions apply.

A = diameter of shaft.

P = (see Fig. 30).

$a = 2\frac{1}{16} A + 1\frac{3}{4}"$.

$b = 1\frac{1}{4} A + 1\frac{3}{4}"$.

$c = 2 P$.

$e = P + \frac{1}{16}"$.

$f = 1\frac{3}{4} P$.

$g = \frac{1}{10} A + \frac{1}{4}"$.

$h = \frac{3}{4} g$.

$k = \frac{1}{3} c + \frac{1}{8}"$.

m = diameter of bolts = g .

$n = 3 g$.

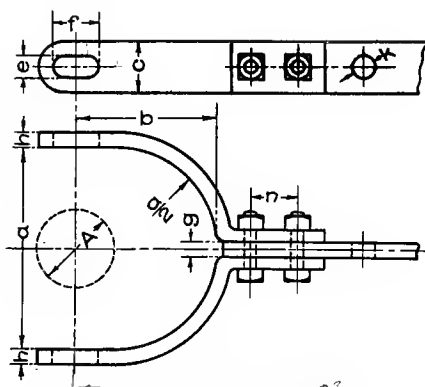


FIG. 31.

Friction Clutches are made in too many forms and of construction too complex for discussion here but their general characteristics are illustrated in Fig. 32. A collar sliding on the shaft controls the levers which operate the friction grip. Sliding of this collar may be effected by a hand lever like that illustrated in Fig. 29, if the clutch is small. For large clutches, a hand wheel and gears as illustrated in Fig. 33 are used to operate the lever. These clutches

are largely used where the connection has to be made when the driving shaft is in motion at a considerable speed, since they avoid

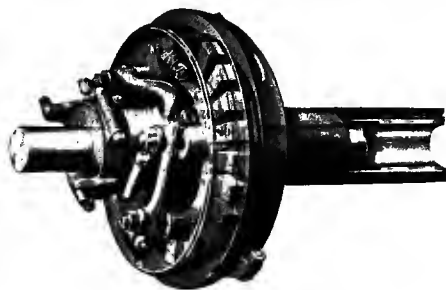


FIG. 32.

the sudden shock of a positive connection, but they have the disadvantage of requiring good adjustment to prevent slipping under load.

22. **Collars.**—A shaft is prevented from endwise motion through the supporting bearings by means of collars, Fig. 34, which are clamped to the shaft by means of set screws. The finished sides



FIG. 33.

of these collars bear against the finished ends of the bearings. Formerly plain collars of rectangular section with standard set screws were used. These were dangerous because of the projecting set screw heads which caught the clothes of workmen, winding

them about the shaft. Many states now have laws prohibiting their use. The common headless set screws which are slotted so as to be tightened with a screw-driver were too weak to be satisfactory when used with the plain collars but the introduction of



FIG. 34.

safety set screws, Fig. 13, page 23, has removed this difficulty. Such screws used with the plain collar should be short enough to be entirely beneath the surface of the collar when tightened.

In the safety collars, Fig. 34, the set screw head is surrounded by a ring and the projecting flanges at the sides protect the workmen from contact with this ring as it revolves. Safety collars of

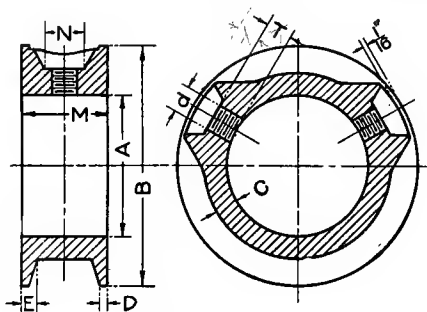


FIG. 35.

the general form shown are made in halves that may be bolted together around the shaft and thus avoid taking down a shaft to put on an additional collar. The following empirical proportions for safety collars were derived from dimensions taken from commercial collars of the form drawn in section in Fig. 35 and sold by one of the leading makers. One low head set screw is used for shafts under 3" in diameter and two for larger sizes.

d = diameter of set screw.

L = length of set screw.

A = diameter of shaft.

B = $A + 2L + d$.

C = $\frac{1}{32}A + \frac{3}{8}$ ".

D = $\frac{1}{48}A + \frac{1}{8}$ ".

E = $\frac{1}{24}A + \frac{1}{8}$ ", but not to exceed $D + \frac{1}{8}$ ".

T = $\frac{4}{5}L$.

	One Set Screw.	Two Set Screws.
d	$\frac{5}{32}A + \frac{3}{16}$ ".	$\frac{1}{16}A + \frac{3}{8}$ ", but not to exceed $\frac{9}{64}A$.
L	$d + \frac{1}{8}$ ".	$2d - \frac{1}{4}$ ".
M	$2d + \frac{5}{8}$ ".	$3\frac{1}{2}d + \frac{1}{4}$ ".
N	$2d + \frac{1}{8}$ ".	$2d$, but not to exceed $d + \frac{3}{4}$ ".

CHAPTER V.

SHAFT FIXTURES.

23. General Nature.—Under the head of shaft fixtures are included all of those fixed parts by means of which a shaft is sustained in its proper position with regard to the building in which it is located. These fixtures may be conveniently divided into the **bearings**, which are actually in contact with the shaft, and the **bearing supports** intermediate between the bearing and the posts, walls, or floor timbers, which furnish the ultimate support for the shaft.

24. Purpose and Qualities of Bearings.—The purpose of a bearing is to support a shaft and to constrain it to revolve about its own axis while the bearing remains attached to some stationary body. For this reason it must present a polished, well lubricated inner surface for contact with the surface of the shaft. This surface must be of such material as to cause the least possible damage to the shaft in case of failure of the lubrication and, at the same time, to resist wear when properly lubricated. Cast iron furnishes a fairly good surface as long as it is well lubricated but, in case the lubricant fails, it does great damage to the shaft because of its superior hardness. To save the shaft the bearing metal should be the softer. Materials, such as brass, bronze, or babbitt metal, which combine this softness with the requisite wearing qualities are too expensive to use for the construction of the whole of the bearing and are used simply as a lining for the inner surface, the frame or remainder of the bearing being of cast iron. In case of damage these linings are easily replaced.

When the lining is of brass or bronze it is usually machined to fit a correspondingly machined surface of the cast iron frame. Linings of babbitt metal, owing to its low melting point, are cast in the frame which is already completed in other ways. There are grooves or holes, called **anchorages**, in the inner surface of the frame, into which the metal sets and is prevented from rotating or sliding axially. In the cheaper bearings a short piece of shafting

of the proper diameter is carefully centered in the frame with close fitting collars at the ends and the babbitt metal poured around the shafting. The shrinkage of the metal as it cools tends to draw such linings away from the frame. The better bearings are poured with a surplus of metal which is afterward hammered into the anchorages and the surface machined to size.

25. Forms of Bearings.—The varying positions of the shaft, methods of supporting the bearings, and methods of supplying the



FIG. 36.

lubricant to the bearing surfaces have resulted in so many forms of bearings that but the briefest mention may be made of them in this book. Usually the shaft is horizontal and the bearing is of some such form as shown in part section in Fig. 36(a). If it be vertical the general form changes to that of Fig. 36(b). Since the set screws of the collars described in article 22 cannot be depended upon to support the weight of the shaft, there must also be provided for a vertical shaft a special form of bearing, known as a **step bearing**, Fig. 36(c), in which the lower end of the shaft rests. Bearings may be solid, as in Fig. 36(b), requiring to be slipped on over the end of the shaft. In most cases, however, they are split along the center line of the shaft, Fig. 36(a), so that it may easily be removed. This also provides opportunity to take up looseness due to wear. These parts are designated as the **cap** and the **base** and they are held together by two or more bolts, known as the **cap bolts**.

A bearing may be supported from beneath, requiring a flat bottom for the base, Fig. 36(a); from the side, requiring a vertical side on the base, Fig. 36(d); or it may be suspended on pivots at the center, Fig. 36(e). The lubricant may be supplied through an oil hole, Fig. 36(d); it may be carried from an oil reservoir by a wick, which is pressed against the shaft, Fig. 36(a); or it may be carried up to the top of the shaft from a reservoir beneath by means of rings or chains, Fig. 36(e), which rest upon and revolve with the shaft, while dipping into the oil in the reservoir.

26. Adjustments of Bearings.—In order that a shaft may run properly its axis must be as near to a straight line as it is practicable to obtain. To accomplish this straightening or **alignment** of the shaft provision must be made so that the supporting bearings may be adjusted either vertically or horizontally or in both directions in a plane perpendicular to the axis of the shaft. One of these adjustments, the horizontal, in the case of Fig. 36(a) and the vertical in Fig. 36(b) and Fig. 36(d), is usually provided for by elongation of the holes in the base through which it is bolted to a support. These bolts may be designated as the **holding bolts** to distinguish them from the cap bolts. The remaining adjustment, or, in the case of Fig. 36(e), both adjustments, should be provided in the support for the bearing. The making of these adjustments is called **aligning the shaft**.

In split bearings provision is made to prevent the shaft from becoming loose as the lining of the bearing wears away. When the lining is new thin strips of metal or hard pressed paper, called **liners** or **shims**, are placed on each side of the shaft between the cap and base so that the cap bolts may be tightened firmly without pinching the shaft. These liners are then removed one at a time

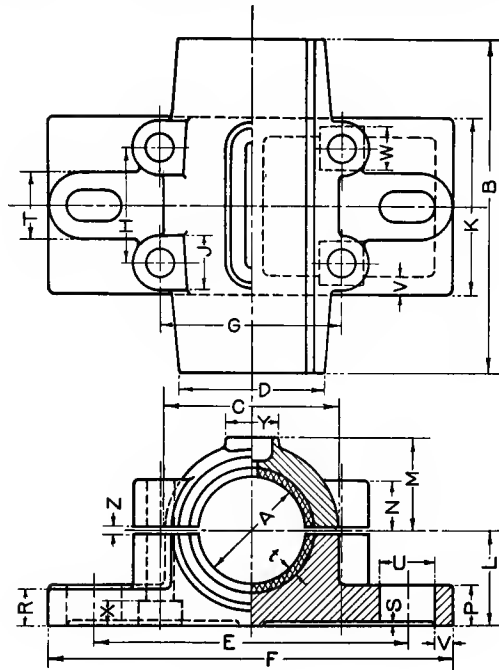


FIG. 37.

as the wear progresses until all have been removed, when a new lining is put in and the process repeated. In split bearings the edges of the linings along the division should be chamfered so that the lubricant may not be scraped off and forced out at the joint but may be drawn around with the shaft. This chamfering should not extend to the ends of the bearing lest it furnish a means of escape for the lubricant.

Endwise adjustment of horizontal shafts is provided by means of setting the collars (see article 22 on page 43) which bear against

the finished ends of the bearings. In vertical shafts this adjustment is made by raising or lowering the step bearing.

27. Proportions for Babbitted Bearings.—The following empirical equations were derived from measurements taken from a simple form of commercial babbitted shaft bearing, designed to be supported from beneath and having oil hole lubrication as shown in the key drawing, Fig. 37. In small bearings, for shafts less than $2\frac{3}{4}$ " in diameter, two cap bolts are used and holes drilled for babbitt anchorages. In larger sizes four cap bolts are used and dovetail grooves cored for the anchorages. To prevent endwise displacement of the babbitt these anchorages should not extend to the ends of the bearing. The cap bolts and holding bolts are provided with hexagon nuts.

A = diameter of shaft.

d = diameter of holding bolts = $\frac{1}{8}A + \frac{7}{16}"$ but not to exceed $\frac{1}{2}A$.

d₁ = diameter of cap bolts = $\frac{3}{4}d$.

t = $\frac{1}{16}A + \frac{1}{16}"$, but not to exceed $\frac{1}{32}A + \frac{3}{16}"$, or $\frac{1}{2}"$.

B = length of bore = 3 A.

C = $1\frac{5}{16}A + 2t + \frac{1}{4}"$.

D = C - $\frac{1}{8}A$.

E = $1\frac{3}{4}A + 4"$ (4 cap bolts),

= G + J + 3d + $\frac{1}{4}A + \frac{1}{2}"$ (2 cap bolts).

F = E + 3d + $\frac{1}{2}"$.

G = $1\frac{1}{8}A + 2t + d_1 + \frac{1}{4}"$.

H = $\frac{5}{8}A + 1\frac{1}{2}"$, but not to exceed A.

J = 2 d₁ + $\frac{1}{8}"$.

K = H + J + $\frac{1}{16}A + \frac{1}{8}"$ (4 cap bolts),

= $1\frac{1}{2}A + \frac{1}{2}"$ (2 cap bolts).

L = $\frac{5}{6}A$, but not less than $\frac{1}{2}C + \frac{1}{8}"$.

M = $\frac{1}{2}C - \frac{1}{32}A + \frac{1}{2}"$, but not to exceed A.

N = $\frac{7}{16}A + \frac{5}{16}"$ (approx.).

P = $\frac{1}{3}A$, but not less than $\frac{1}{4}A + \frac{1}{2}"$.

R = P - $\frac{1}{16}A$.

S = $\frac{1}{8}A - \frac{1}{8}"$.

$$T = 2d + \frac{1}{2}''.$$

$$U = 2d + \frac{1}{8}''.$$

$$V = \frac{1}{8}A + \frac{1}{4}''.$$

$$W = 1\frac{1}{2}d_1 + \frac{1}{4}''.$$

$$X = \text{not less than } d_1 + \frac{1}{16}''.$$

$$Y = \frac{3}{8}A + \frac{1}{2}''.$$

$$Z = \frac{1}{2}A.$$

In order that the oil hole in the top of the cap may be made without machining or coring it is made large enough to mold in

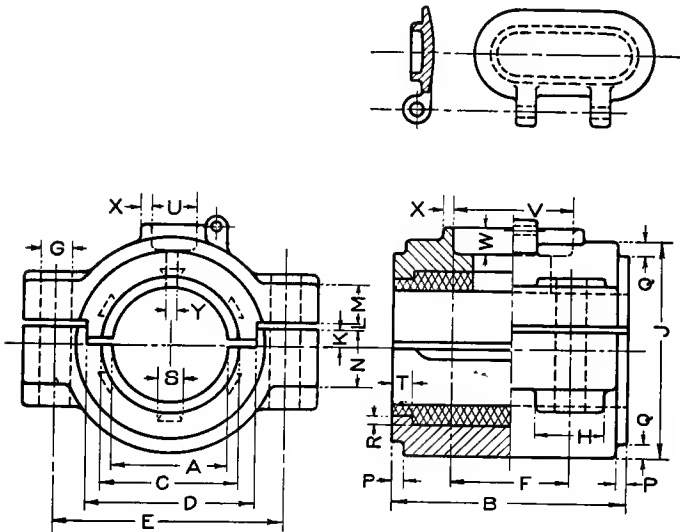


FIG. 38.

green sand. The oil is prevented from flowing out around the shaft too freely by filling the oil well with cotton waste or other absorbent material.

The general proportions given below, which apply to the key drawing, Fig. 38, are for the type of babbitted bearing suitable for machine frames. The grease cup shown may be replaced by any suitable form of oiling device without other changes in the bearing. Through bolts should be used whenever possible. If the bearing

is so placed that this is not possible, studs or tap-bolts may be substituted. The offset in the division between the cap and the base is machined to an accurate fit and prevents sidewise displacement of the cap. The cover to the oil well being a separate casting may be relatively thin.

A = diameter of bore.

B = length of bore = A to 4 A to suit conditions.

t = thickness of babbitt = $\frac{1}{16}A + \frac{1}{8}"$, but not to exceed $\frac{1}{32}A + \frac{1}{4}"$.

C = A + 2 t.

D = $1\frac{3}{8}A + \frac{1}{4}"$.

d = diameter of bolts = $\frac{3}{16}A + \frac{7}{4}"$
(use 4 except for short bearings, then use 2).

E = D + $1\frac{3}{4}d$.

F = $\frac{1}{2}B$.

G = d + $\frac{1}{8}"$.

H = 2 d + $\frac{1}{4}"$.

J = $1\frac{3}{4}A + \frac{1}{2}"$.

K = $\frac{1}{8}A + \frac{1}{8}"$.

L = $\frac{1}{2}t$.

M = $\frac{5}{16}A + \frac{1}{8}"$.

N = M + K.

P = $\frac{1}{8}A$ or to suit.

Q = $\frac{1}{8}A$ or to suit.

R = $\frac{3}{4}t$.

S = 3 t - $\frac{1}{4}"$.

T = 2 t - $\frac{1}{8}"$.

U = $\frac{5}{16}A + \frac{5}{16}"$.

V = $\frac{1}{2}B$.

W = $\frac{5}{8}U$.

X = $\frac{1}{16}A + \frac{1}{16}"$.

Y = $\frac{3}{64}A + \frac{3}{16}"$.

28. Quarter-box Bearings.—These bearings get their name from the fact that the bearing surface or box is divided into four

parts or "quarters," Fig. 39, each of which may be moved up against the journal by means of independent adjusting wedges or screws. By this means any wear that may have occurred can be taken up in a more nearly correct manner than could be done with

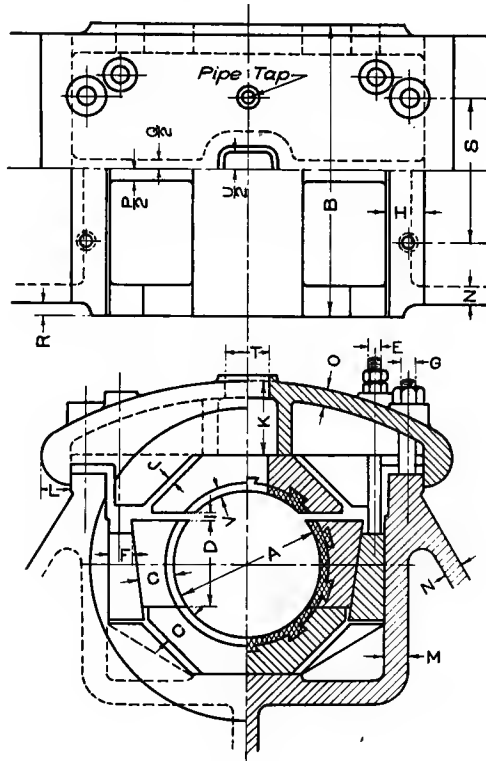


FIG. 39.

bearings divided into but two adjustable parts. When the wear in a bearing is due to a resultant pressure in a single direction, such as the weight of the supported shaft alone, a two-part bearing will provide the only needed adjustment. When, however, the direction of this resultant pressure changes during the rotation of the shaft it becomes necessary to provide a more complete adjustment to take up the wear, and also to keep the shaft in true alignment.

Bearings of this type are much used for steam engines and are usually of quite large dimensions. They are not purchasable alone. Either the base is a part of the engine frame or it is a pedestal designed especially to be attached to the engine frame or to its foundation. The base is recessed to receive the parts of the box and the adjusting wedges. A strip in the center of the bottom of this recess is machined to receive the bottom section of the box, which may or may not be provided with a vertical adjustment by liners or a wedge. A portion of each side of the recess is machined to support the adjusting wedges which move the box and shaft horizontally. This adjustment is necessary to keep the correct distance between the center line of the shaft and the center of the cylinder. Each of these wedges is drawn up by means of two studs passing through the cap which, since they cannot be drawn tight without binding the shaft, are kept in position by means of jam nuts. The motion of these wedges should be sufficient to take up wear equal to one-half the thickness of the babbitt on each of the side sections of the box. The cap and base are machined to an accurate fit at the sides and a strip in the center of the under side of the cap is machined to bear on the top section of the box. The cap is drawn down by means of studs set in the top of the base until the top section of the box bears firmly on liners placed between it and the side sections. Each of the sections of the box is machined for contact with the adjacent sections.

These boxes when worn or in case of accidental over-heating have to be removed to be relined with babbitt. It is, therefore, desirable that it should be possible, by loosening the cap and blocking up the shaft, to slide them out and replace them without removing the shaft from position. Where this sliding is endwise it is necessary to attach a circular plate to the frame or pedestal and around the shaft or to provide some other means to keep them from working out from the vibration of the engine while running.

Under normal conditions the lubricant is oil fed either from large oil cups or piped to the bearing from a tank. In addition there is usually a provision for an emergency lubrication by packing a recess in the cap with some solid grease that will melt and flow into the bearing should the temperature rise above the normal.

There are many designs of quarter-box bearings, that of which the description has been given and to which the following empirical proportions apply, being one of the simplest. Some of the variations from this design are, use of a single adjusting wedge; substitution of set screws for the adjusting wedges; adjustment of the lower box by means of a wedge; adjustment of the upper box by means of set screws; and making the bearing self-oiling by providing an oil reservoir and oiling chains running in suitable channels in the boxes.

A = diameter of bore.

B = length of bore = $1\frac{1}{2}$ A to $2\frac{1}{2}$ A.

C = 0.225 A + $\frac{1}{4}$ ".

D = $\frac{3}{5}$ A.

E = diameter of wedge bolts = $\frac{1}{15}$ A + $\frac{1}{4}$ ".
(Core holes in cap $\frac{1}{4}$ " to $\frac{1}{2}$ " larger.)

F = minimum thickness of wedge = 2 E.
(Taper = $1\frac{1}{2}$ " in 12 ".)

G = diameter of cap bolts = $\frac{1}{10}$ A + $\frac{1}{4}$ ".
(Core holes in cap $\frac{1}{4}$ " to $\frac{1}{2}$ " larger.)

H = 2 G + $\frac{1}{4}$ ".

I = $\frac{1}{32}$ A + $\frac{1}{4}$ ".

J = $\frac{1}{40}$ A + $\frac{1}{4}$ ".

K = 0.45 A + $\frac{1}{2}$ ".

L = $\frac{1}{6}$ A + $\frac{1}{4}$ ".

M = $\frac{3}{32}$ A + $\frac{7}{8}$ ".

N = $\frac{3}{4}$ M.

O = $\frac{5}{8}$ M.

P = $\frac{7}{8}$ M.

Q = $\frac{3}{4}$ M.

R = $\frac{3}{8}$ M.

S = $\frac{1}{2}$ B.

T = $\frac{1}{4}$ A + $\frac{3}{4}$ ".

U = $\frac{3}{4}$ T.

V = $\frac{1}{48}$ A + $\frac{1}{4}$ ".

29. **Bearing Supports.**—Bearings may be attached directly to the top or to the under side of floor timbers, to posts, or to walls. It rarely occurs, however, that the shaft is brought into the desired position when so fastened. Interposing an intermediate member

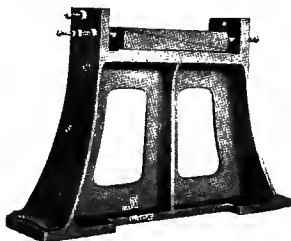


FIG. 40.

not only removes the shaft farther from the wall, floor, or ceiling but may also provide a convenient adjustment for aligning the shaft. These intermediate members vary in size and proportion according to the diameter of the shaft with which they are used, and with the distance of the shaft from the floor, wall, or ceiling from which it is supported.



FIG. 41.

30. **Stands and Base Plates.**—When the shaft is to be supported from beneath, the intermediate member may have the general form of the **floor stand**, Fig. 40. The stock sizes vary by 6" in nominal height up to 42". This height may, in each size, be varied a small amount (usually less than an inch) by means of the adjusting wedges. These are operated by means of the set screws and secured by the jam nuts shown at the sides. When it is desirable that the elevation of the shaft from the floor shall be small a **base plate**, Fig. 41, may be used in the place of the stand. This may or may not have the provision for vertical adjustment.

31. **Wall Brackets.**—When the bearing is to be supported from a wall or post it may be placed upon a wall bracket as illustrated



FIG. 42.

in Fig. 42. These brackets are fastened to a wall by means of through bolts, or to wooden posts by means of hanger screws, or

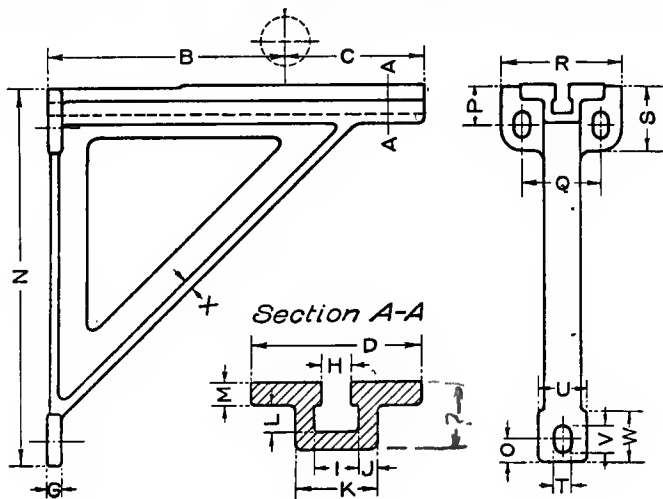


FIG. 43.

lag screws. The holes in the bracket for these fastenings are elongated vertically to provide an adjustment for aligning the shaft. In order to reduce the number of stock sizes of these brackets, the manufacturers proportion them so that the distance of the shaft from the wall, called the **extension** of the bracket, may be varied a sufficient number of inches either way from the nominal value so as to meet the extensions of the adjacent sizes. For the same reason the proportions for each nominal extension of bracket, which vary with the diameter of the shaft to be supported, are taken suitable for a certain maximum diameter of shaft and that bracket used for the several diameters of shaft next smaller. These considerations determine the values of C, D and E in the following proportions which were taken from the catalogue of a prominent maker and which apply to the key drawing, Fig. 43. In these proportions, A is the diameter of the shaft to be supported; B is the nominal extension of the bracket; E is the diameter of the holding bolts for the bearing, the square heads of which are placed in, and held from turning by the sides of, the grooves in top of the bracket.

A	B	C	D	E	F	G	A	B	C	D	E	F	G
1 $\frac{3}{4}$	12	8	4 $\frac{1}{4}$	3 $\frac{3}{4}$	3 $\frac{3}{4}$	3 $\frac{3}{4}$	3 $\frac{3}{4}$	12	11	6 $\frac{3}{4}$	1	1	1
	18	8	4 $\frac{1}{4}$	3 $\frac{3}{4}$	3 $\frac{3}{4}$	3 $\frac{3}{4}$		18	11	6 $\frac{3}{4}$	1	1	1 $\frac{1}{8}$
to	24	8	4 $\frac{1}{4}$	3 $\frac{3}{4}$	1	1 $\frac{1}{8}$	to	24	11	6 $\frac{3}{4}$	1	1 $\frac{1}{8}$	1 $\frac{1}{4}$
	30	8	4 $\frac{1}{4}$	3 $\frac{3}{4}$		1 $\frac{1}{8}$		30	11	6 $\frac{3}{4}$	1	1 $\frac{1}{8}$	1 $\frac{3}{8}$
2 $\frac{1}{2}$	36	8	4 $\frac{1}{4}$	3 $\frac{3}{4}$	1	1 $\frac{1}{4}$	4 $\frac{1}{2}$	36	11	6 $\frac{3}{4}$	1	1 $\frac{1}{4}$	1 $\frac{1}{2}$
2 $\frac{3}{4}$	12	10	5 $\frac{3}{4}$	7 $\frac{7}{8}$	7 $\frac{7}{8}$	7 $\frac{7}{8}$	4 $\frac{3}{4}$						
	18	10	5 $\frac{3}{4}$	7 $\frac{7}{8}$	1	1		18	12	7 $\frac{1}{2}$	1 $\frac{1}{8}$	1 $\frac{1}{8}$	1 $\frac{1}{4}$
to	24	10	5 $\frac{3}{4}$	7 $\frac{7}{8}$		1 $\frac{1}{8}$	to	24	12	7 $\frac{1}{2}$	1 $\frac{1}{8}$	1 $\frac{1}{4}$	1 $\frac{3}{8}$
	30	10	5 $\frac{3}{4}$	7 $\frac{7}{8}$	1	1 $\frac{1}{4}$		30	12	7 $\frac{1}{2}$	1 $\frac{1}{8}$	1 $\frac{1}{4}$	1 $\frac{1}{2}$
3 $\frac{1}{2}$	36	10	5 $\frac{3}{4}$	7 $\frac{7}{8}$	1 $\frac{1}{8}$	1 $\frac{3}{8}$	5 $\frac{1}{2}$						

$$H = E + \frac{1}{8}''.$$

$$I = 1\frac{1}{2}E + \frac{1}{4}''.$$

$$J = \frac{2}{3}E.$$

$$K = I + 2J.$$

$$L = E + \frac{1}{16}''.$$

$$M = E - \frac{1}{8}''.$$

$$N = B + C.$$

$$O = 2F.$$

$$P = 2F + M.$$

$$Q = K + 3F.$$

$$R = Q + 3F.$$

$$S = P + 2F.$$

$$T = F + \frac{1}{8}''.$$

$$U = \frac{3}{4}D.$$

$$V = 2F.$$

$$W = 4F.$$

$$X = \frac{1}{2}G.$$

32. Wall Box Frames.—When it is necessary to have a bearing where a shaft passes through a wall, a wall box frame, Fig. 44, is set in the wall to support the bearing, which is fastened to it by means of bolts or studs. This frame may be provided with wedges for

vertical alignment of the shaft as shown in Fig. 44, or the bearing may be bolted directly to the bottom of the frame as indicated in

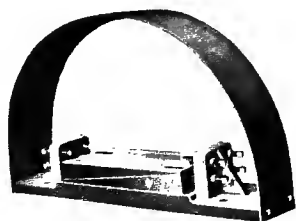


FIG. 44.

the key drawing, Fig. 45, to which the following description and empirical proportions apply. The raised strips on the inside of the bottom of the frame are to provide the necessary finished surface for contact with the finished portion of the bottom surface of the base of the bearing without machining the entire surface. Since each size of frame takes several sizes of bearings the width of these finished strips must be sufficient to permit the finished portions extending across the ends and center of any size of bearing, within the range used with this frame, to rest wholly on the strips when at

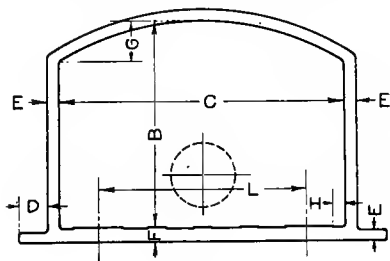


FIG. 45.

the extreme points of sidewise adjustment in either direction. Should there be a considerable space between the outer and center strips in any frame additional intermediate strips may be placed there to furnish a support to the bearing along the edge of the base.

The sizes of these frames are designated by numbers, each number serving for sizes of bearings as follows:

Frame Number.	1	2	3	4	5	6
Diameter of Shaft.	$1\frac{3}{4}$ to $2\frac{1}{2}$	$2\frac{3}{4}$ to $3\frac{1}{2}$	$3\frac{3}{4}$ to $4\frac{1}{2}$	$4\frac{3}{4}$ to $5\frac{1}{2}$	6 to $6\frac{1}{2}$	7 to $7\frac{1}{2}$

In general they must be designed with reference to the particular form of bearing they are to support. The proportions here given were determined for use with the babbitted bearing described in article 27 on page 50.

A = maximum diameter of shaft to be used in frame.

B = $2\frac{1}{8}A + 2\frac{1}{2}"$.

C = $2\frac{3}{8}A + 7"$.

D = $\frac{3}{8}A$.

E = $\frac{1}{16}A + \frac{11}{32}"$.

F = E + $\frac{1}{4}"$.

G = $\frac{1}{8}B$.

H = $\frac{1}{8}A$.

K = $\frac{1}{2}C$ = width of frame.

L = distance between holding bolts for bearing (see article 27).

33. Hangers.—When the bearing is to be supported from above, the intermediate member takes the form of a **drop hanger**, Fig. 46, which receives a bearing of the type shown in Fig. 36(e) on page

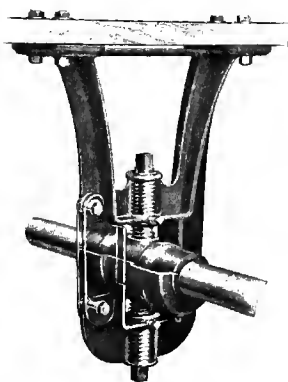


FIG. 46.

47. Horizontal adjustment is provided by the elongated holes for the bolts at the top of the hanger and vertical adjustment is secured from the pivot screws shown above and below the bearing. The drop hangers are carried in stock with different amounts of drop increasing by 2" intervals from 8" up to 24" and by 6"

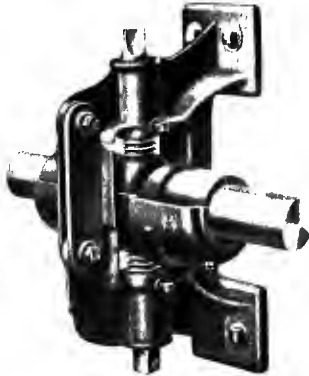


FIG. 47.

intervals from that to 36" except that shafts above 3" in diameter require more than the minimum drop of 8". An adaptation of these hangers for attaching to a post is shown in Fig. 47. These **post hangers** provide no variation of the distance of the shaft from the post. At the left of each figure may be seen the brace links which may be removed so that the shaft may be dismantled without removing the hanger from its fastenings.

CHAPTER VI.

TRANSMISSION MEMBERS.

34. General Statement.—In this chapter are considered some of those machine parts which are usually attached to revolving shafts. These include such members as pulleys, gears, cams, etc. Their adjacent members such as belts and cam followers are briefly mentioned. No attempt has been made to carry the discussion beyond the information needed for the drawings of a brief course in Empirical Design.

35. Pulleys.—The use of pulleys either keyed to shafts for the purpose of transmitting power or running loose on shafts for the purpose of guiding or supporting belts is too common to require any description. Pulleys are usually made of cast iron, of wood, of stamped steel or of combinations of these materials. They are made both solid and split. Split pulleys consist of two equal halves bolted together at hub and rim. Otherwise the proportions are the same as for solid pulleys. Driving pulleys and those under heavy loads should be keyed to the shaft. Those under light loads may be secured to the shaft by means of set screws or the compression of bushings.

There is a wide variation in the diameters of shafts upon which a pulley of any nominal size (outside diameter) may be used. The hubs are made large enough for the largest probable diameter of shaft. In solid pulleys, and in split pulleys which are to be keyed to the shaft, the hubs are bored and the keyways cut to order before shipping from the factory. Split pulleys which are to be secured by the compression of bushings have their hubs already bored to receive the largest diameter of shaft. For shafts of smaller diameters split bushings are supplied to make the necessary reduction. Such pulleys and a supply of bushings are usually carried in stock by dealers.

The stock sizes vary to some small extent with different makers. In general they vary by 1" from 6" to 36" in diameter and by 2" from 36" up to 144". The widths of face vary by 1" from 3" up to

12" and by 2" from 12" up to 60". The minimum stock width increases from 3" for pulleys 36" and less in diameter, up to 12" for 112" and over in diameter. The maximum width of face increases from 12" for pulleys 6" and 7" in diameter, up to 60" for pulleys 96" and over in diameter. When the hubs are bored and the keyways cut to order the width of face may be reduced from the nearest stock size to any desired width at a slight added expense. Obviously any change from a stock diameter will require building entirely to order at a correspondingly increased cost.

The arms of pulleys are usually elliptical in cross section. The arms of very large cast iron pulleys are tapered both in width and

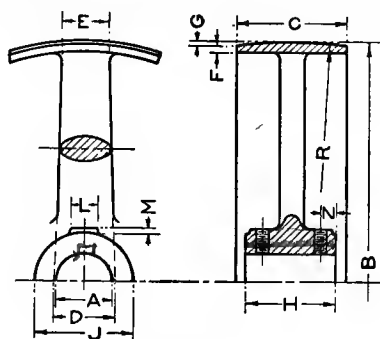


FIG. 48.

thickness, those of smaller pulleys in width only. The amount of taper in width varies from $\frac{1}{4}$ " to $\frac{3}{8}$ " per foot per side, and in thickness from $\frac{1}{8}$ " to $\frac{3}{16}$ " per foot per side. The thickness of pulley arms is made from 0.4 to 0.5 the width. The wider pulleys are made with a double set of arms to sustain the rim. Some pulleys not so wide in proportion to their diameters are made with either single or double sets of arms. In addition to the bore of the hub and the face of the rim the ends of the hub and the edges of the rim are finished.

The following proportions for cast iron pulleys having six arms apply to the key drawing, Fig. 48, when all dimensions are in inches.

A = diameter of shaft.

B = diameter of pulley.

C = width of face = $1\frac{1}{8}$ width of belt.

$D = \frac{3}{8} \sqrt[3]{B \times C}$ for single belts,

$= \frac{7}{16} \sqrt[3]{B \times C}$ for double belts.

$E = D - \frac{1}{48}B$ to $D - \frac{1}{32}B$.

$F = \frac{1}{8}E$ (taken to next larger $\frac{1}{32}$ " for values less than $\frac{1}{2}$ ").

$G = \frac{1}{4}$ " per foot width of face. (This is an average value. The crown is made greater on narrow pulleys and less on wide ones.)

$H = \frac{3}{4}C$, (This is an average value. H should be greater for loose than for tight pulleys.)

$= \frac{3}{8}C + \frac{1}{16}B + 1"$. (This gives good values for tight pulleys.)

$J = A + \frac{2}{3}D$ but not to exceed $1\frac{3}{4}A + \frac{1}{2}"$.

K = diameter of set screw

$= \frac{5}{16}(J - A) + \frac{3}{16}"$ when bearing on shaft,

$=$ width of key when bearing on key.

$L = 2K$.

$M = \frac{1}{2}K$.

$N = 1\frac{1}{2}K$.

The following equations give satisfactory radii for drawing the several curved surfaces but should not be given as dimensions.

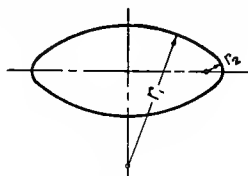


FIG. 49.

For the crown of the pulley $R = C^2 \div 8G$. For the ellipse of the arms, Fig. 49, $r_1 = \frac{3}{4}$ the major axis. r_2 is found by trial to pass through the extremities of the major axis and tangent to the arcs of radius r_1 .

36. Belts.—The discussion in this article will be confined to belts of flat cross section such as would be used on pulleys having rims of the general form shown in Fig. 48. Belts may be made of leather, of cotton, of rubber, or of combinations of these materials.

The ends of the belt are fastened together as smoothly as practicable, forming a closed loop which is passed tightly over the faces of two or more pulleys. Its ability to transmit power is dependent upon the tightness of the belt on the driving and driven pulleys, upon the friction between the belt and the pulleys, and upon the area of cross section of the belt. In commercial belting the stock sizes of cross sections vary both in width and in thickness,

Leather Belting.—The different thicknesses of leather belting are obtained by cementing together two or more thicknesses of leather. Such belting is known as single, double, triple and quadruple leather belting according to the number of thicknesses



FIG. 50.

used. Single belting, $\frac{3}{16}$ " to $\frac{1}{4}$ " thick, and double belting, $\frac{5}{16}$ " to $\frac{7}{16}$ " thick, are the thicknesses commonly used. The standard widths of leather belting are,

$\frac{1}{2}$ " to 1" varying by $\frac{1}{8}$ ";	7" to 28" varying by 1";
1" to 4" varying by $\frac{1}{4}$ ";	28" to 40" varying by 2";
4" to 7" varying by $\frac{1}{2}$ ";	40" to 72" varying by 4".

The following empirical equations give good values for the power which may be transmitted by leather belts having effective tensions of 38 pounds and 60 pounds per inch of width for single and double belts respectively, at speeds not exceeding 1,000 feet per minute.

D = diameter of pulley in inches.

N = revolutions per minute of pulley.

W = width of belt in inches.

H. P. = horse power transmitted

$$= \frac{W D N}{3,300} \text{ for single belts,}$$

$$= \frac{W D N}{2,100} \text{ for double belts.}$$

37. Handwheels.—A handwheel is used in the place of a crank or wrench in places where it is desirable to be able to grasp with the

same ease and force in all phases of a rotation. The handwheel consists of a hub and spokes of the form usual to pulleys, and a rim of such form as to readily fit the hand. The rim is most frequently

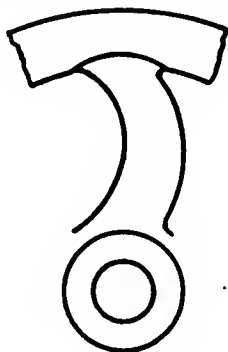


FIG. 51.

circular in section or modified as in Fig. 50(a) to afford an easier grip for the larger sizes. The U. S. Navy uses a rectangular form with the corners slightly rounded in its smaller sizes and modified to the form of Fig. 50(b) in its larger sizes. Small sizes may be finished all over but in the larger sizes the rim and ends of the hub alone are finished. The spokes or arms are usually straight but may be curved as in Fig. 51 to relieve stresses due to shrinkage in

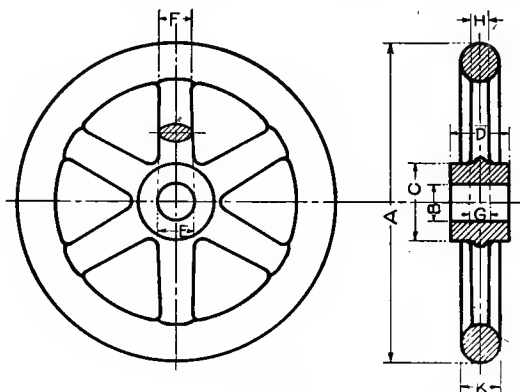


FIG. 52.

casting. The most common cross section is in the form of an ellipse.

Handwheels are not carried in stock and no standard proportions can be given. The nominal size is the outside diameter of the rim. Table XIV gives good values for the dimensions shown in the key drawing, Fig. 52, for the 6" and 16" sizes. Values for these dimensions for other sizes from 4" to 24" may be obtained by the graphical method described in article 3 on page 5, using a straight line for the curve in each case.

TABLE XIV.
PROPORTIONS FOR CAST IRON HANDWHEELS.

A	B	C	D	E	F	G	H	K
6	$\frac{5}{8}$	$1\frac{1}{4}$	$\frac{15}{16}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{5}{8}$
16	$1\frac{1}{4}$	$2\frac{1}{2}$	$1\frac{7}{8}$	$1\frac{1}{4}$	1	$\frac{5}{8}$	$\frac{1}{2}$	$1\frac{1}{4}$

38. General Nature and Properties of Gears.—A gear consists primarily of a hub and arms, similar to those for pulleys, which

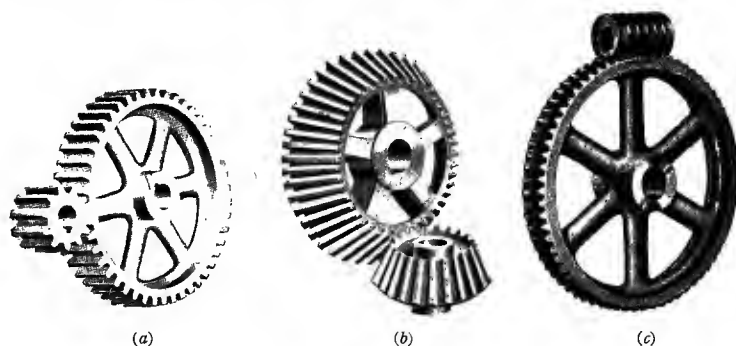


FIG. 53.

support a rim upon the surface of which teeth are formed. These teeth interlock with the teeth of a mating gear, Fig. 53, to transmit the power. Where one of these gears has a small number of teeth it is known as a **pinion**. There is an imaginary smooth surface passing through the teeth of each gear at approximately their mid-height which rolls upon the corresponding surface of the mat-

ing gear. These imaginary surfaces are called the **pitch surfaces** of the gears. Gears may be classified in a general way according to the forms of their pitch surfaces. **Spur gears**, Fig. 53(a), have cylinders for pitch surfaces; **bevel gears**, Fig. 53(b), have conical frustums for pitch surfaces; and **worm gears**, Fig. 53(c), have pitch surfaces of double curvature. A pair of equal bevel gears with shafts perpendicular are called **mitre gears**.

A cross section taken through a cylindrical or a conical pitch surface is a circle, called the **pitch circle** or **pitch line**. Its diameter is called the **pitch diameter**. For bevel gears the pitch circle is taken at the larger end of the frustum. The linear distance in inches from center to center of adjacent teeth, measured along the arc of the pitch circle, is known as **circular pitch**. The ratio of the number of teeth to the diameter of the pitch circle in inches is called **diametral pitch**. The following relations are obvious from the above definitions:

D = pitch diameter in inches,

N = number of teeth,

$$p_d = \text{diametral pitch} = \frac{N}{D},$$

$$p_c = \text{circular pitch} = \frac{\pi D}{N} = \frac{\pi}{p_d}.$$

39. Proportions for and Properties of Gear Teeth.—The force which may be exerted by a gear tooth upon its mating tooth is dependent upon its thickness, width, and height, and upon its linear velocity. By the width of the tooth is meant the distance, f , across the face of the gear in Fig. 54. The thickness of a standard gear tooth at the pitch line is one-half the circular pitch (approximate in the case of gears having cast teeth). The height of a standard gear tooth has been empirically fixed at $\frac{2''}{p_d}$ plus an allowance of $\frac{0.157''}{p_d}$ for clearance. Of this height $\frac{1''}{p_d}$ extends outside the pitch line and is called the **addendum**. Experiments by Mr. Wilfred J. Lewis have shown that owing to increasing shock, the allowable working fibre stress in gear teeth decreases as the velocity increases. Table XV gives values of allowable fibre

stresses for cast iron gear teeth for various linear velocities in feet per minute at the pitch line. These values may be increased $2\frac{1}{2}$ times for steel.

TABLE XV.
ALLOWABLE FIBRE STRESSES IN CAST IRON GEAR TEETH.

Velocity. Stress.	0-100 8000	200 6000	300 4800	600 4000	900 3000	1200 2400	1800 2000	2400 1700
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His experiments showed further that increasing the width of the tooth beyond three times the circular pitch was not effective in increasing the strength of the tooth although widths from $2\frac{1}{2} p_c$ to $3\frac{1}{2} p_c$ give good results. These proportions may be summarized in the following equations:

$$a = \text{addendum} = \frac{1''}{p_d};$$

$$h = \text{total height of tooth} = \frac{2.157''}{p_d} = 0.687 p_c;$$

$$f = \text{width of face} = 3 p_c = \frac{9.42''}{p_d};$$

$$v = \text{linear velocity in feet per minute at the pitch line} \\ = \frac{\pi \times D \times \text{R.P.M.}}{12};$$

s = allowable fibre stress in pounds per square inch at velocity, v ;

n = number of teeth on weaker gear;

W = safe load on tooth in pounds

$$= sp_c f \left[0.124 - \frac{0.684}{n} \right] \text{ for } 15^\circ \text{ involute teeth, and for} \\ \text{cycloidal teeth when the diameter of the describing circle} \\ \text{equals the radius of the 12 tooth pinion. (Lewis.)}$$

The teeth of gears are usually machined to exact size and form, in blanks prepared for the purpose. These are called **cut teeth**. If the teeth are large the amount of machining is reduced by casting to approximate form with proper allowance for finishing. It is customary in making drawings for cut gears to draw the blanks without teeth. In cross section views the height of the tooth is indicated on the section of the rim without cross hatching, as

shown in Fig. 54. The number and form of the teeth and their diametral pitch is specified in a note. Some large gears for rough work may have their teeth cast to form as closely as possible and be used without machining. Such gears are said to have **cast teeth**, and drawings for them should show the teeth in detail fully dimensioned. The number of teeth and circular pitch is given in a note.

40. Materials used in Gears.—The material most commonly used for gears is cast iron. Because of the greater amount of wear upon their teeth, pinions to work with cast iron gears are frequently

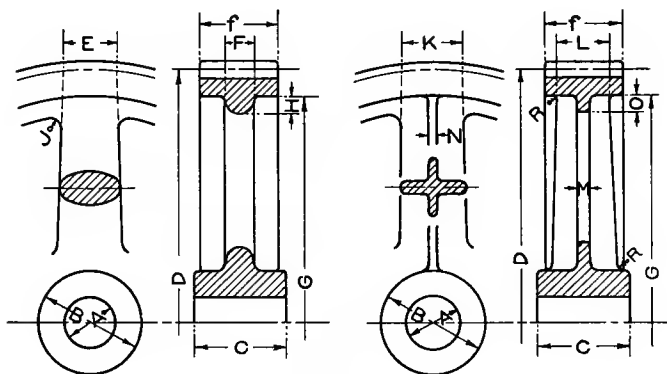


FIG. 54.

made of steel. The same relative equalization of wear may also be obtained by making the teeth of the gear of hard wood mortised into the cast iron rim. Such gears are known as **mortise gears**. When gears are run at high speeds the noise may be greatly decreased by making the pinion of rawhide, fibre, or cloth tightly compressed between cast iron or steel plates at the sides.

41. Proportions for Spur Gears.—Very small spur gears or pinions are made solid without distinction between hub and rim. Gears of sufficient size so that the normal hub and rim would be separated but having less than 36 teeth are made with a continuous web in the place of arms. Gears having from 36 to 60 teeth have four arms. Those having more than 60 teeth should have six arms except in the case of very large gears which may have from eight

to twelve arms. The usual cross sections for the arms of spur gears are elliptical, Fig. 54(a), or in the shape of a cross, Fig. 54(b). These arms are tapered $\frac{3}{8}$ " per foot per side in width and the thickness of elliptical arms is usually made one-half of the width. In small gears the thickness of elliptical arms may be made uniform at an average value.

The following proportions are for cast iron spur gears having six arms and apply to the key drawings, Fig. 54, when all dimensions are in inches. In dimensioning drawings for these gears the outside diameters and pitch diameters should be given as decimals, unless the exact value can be otherwise expressed.

A = diameter of bore.

D = pitch diameter.

p_d = diametral pitch.

p_c = circular pitch.

f = width of face.

n = number of teeth.

$$B = A + 1.6 p_c + \frac{D}{50} = A + \frac{5}{p_d} + \frac{D}{50} = A + \frac{n + 250}{50 p_d}.$$

$$C = f + \frac{D}{40}.$$

$$E = 2\frac{1}{8} p_c = \frac{6.67''}{p_d} = \text{width of arm at pitch line.}$$

$$F = \frac{1}{2} E = \text{thickness of arm at pitch line.}$$

$$G = D - 1.735 p_c = D - \frac{5.45''}{p_d} \text{ (taken to next smaller } \frac{1}{8}'' \text{ or } \frac{1}{4}'').$$

$$H = \frac{1}{2} F.$$

$$J = H.$$

$$K = 2.3 p_c = \frac{7.22''}{p_d} = \text{width of arm at pitch line.}$$

$$L = f - 2 R.$$

$$M = \frac{1}{2} p_c = \frac{1.57''}{p_d}.$$

$$N = 0.3 p_c = \frac{0.94''}{p_d}.$$

$$O = \frac{1}{4} K.$$

$$P = O.$$

$$R = M.$$

Taper of arms = $\frac{3}{8}$ " per foot per side in width.

42. Proportions for Bevel Gears.—Bevel gears may be designed for use on shafts intersecting at any desired angle but that angle is usually 90° . Very small bevel gears or pinions may have the hub and rim solid. Gears somewhat larger have a continuous web

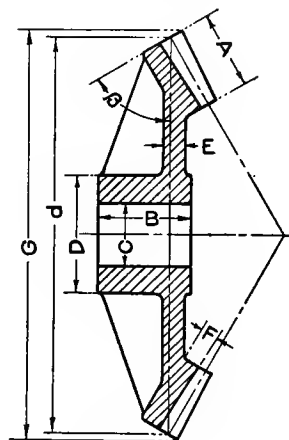


FIG. 55.

joining the rim to the hub with broad stiffening ribs, Fig. 55, to resist the side pressure. In large bevel gears these stiffening ribs are placed opposite the middle of each arm giving it a T section. Bevel gears should be so laid out that as few of the dimensions as possible shall need to be in decimals and at the same time to avoid decimal dimensions on adjacent members. This may be best accomplished by making the distance from the finished surface at the back of the hub to the apex of the conical pitch surface an amount which will not involve a decimal. The distance from this surface to the outer points of the teeth, the outside diameter and the pitch diameter are then the only dimensions which need to be expressed in decimals for cut teeth. If the teeth are to be cast the radii used in laying out the approximate tooth profiles should

also be given as decimals. The angles which the faces of the teeth and the edges of the rim make with a plane perpendicular to the shaft should be given. These angles are called the **face angle** and **edge angle** respectively. The angles and the decimal dimensions should be computed.

The following equations which apply to the key drawing, Fig. 55, give good proportions for cast iron bevel gears when all dimensions are in inches.

d = pitch diameter.

p_c = circular pitch.

p_d = diametral pitch.

A = width of face = $3 p_c - \frac{1}{2}'' = \frac{9.4''}{p_d} - \frac{1}{2}''$, but not to exceed one-third the length of element of pitch cone.

B = length of hub = $\underline{A} + \frac{d}{20}$ (for gears),

$= A + \frac{p_c}{4} = A + \frac{.78''}{p_d}$ (for pinions).

C = bore of hub.

D = diameter of hub = $1\frac{3}{4} C + (\frac{1}{4}'' \text{ to } \frac{1}{2}'')$.

E = thickness of metal in arms or web = $.48 p_c = \frac{1.53''}{p_d}$.

F = distance from bottom of tooth to face of web or arm
 $= .45 p_c = \frac{1.41''}{p_d}$.

(Used only in laying out and not given on drawing.)

G = outside diameter = $d + \frac{2 \cos \beta}{p_d}$.

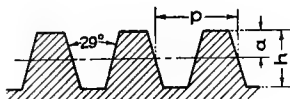


FIG. 56.

43. Worm Gears.—The worm, shown at the top in Fig. 53(c), is the driving member. A cross section through the screw thread of a standard worm is shown in Fig. 56. The following equations

apply to the key drawings, Figs. 56 and 57, and give the usual proportions for steel worms working with cast iron gears. All dimensions are in inches.

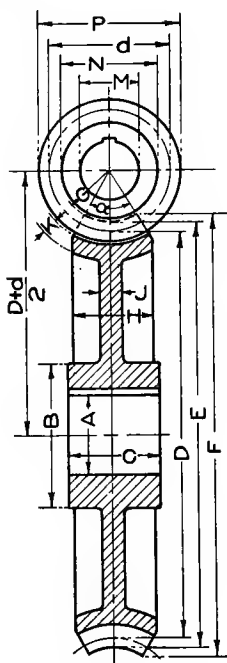


FIG. 57.

d = pitch diameter of worm.

p = linear pitch.

a = addendum = $0.3183 p$.

h = depth of thread = $0.6866 p$.

n = number of threads in worm.

L = lead of worm = $n \times p$.

T = number of teeth in worm wheel.

R = velocity ratio of worm to wheel = $\frac{T}{n}$.

A = bore of wheel.

B = diameter of hub = $1\frac{3}{4} A + (\frac{1}{4}'' \text{ to } \frac{1}{2}'')$.

C = length of hub = $H + \frac{1}{30} D$ or to suit.

D = pitch diameter of wheel.

E = throat diameter of wheel = $D + 2a$.

F = outside diameter of wheel (untrimmed) = $E + 2G (1 - \cos \frac{\alpha}{2})$.

G = throat radius of wheel = $\frac{1}{2}P - 2a$.

H = width of face of wheel = $P \sin \frac{\alpha}{2} + (\frac{1}{8}'' \text{ to } \frac{1}{2}'')$.

J = thickness of web = $0.48 p$.

K = thickness of rim = $0.48 p$.

M = bore of worm in inches.

N = root diameter of worm = $P - 2h$.

P = outside diameter of worm.

Q = minimum length of thread on worm = $\sqrt{E^2 - (E - 4a)^2}$.

α = face angle of wheel.

β = gashing angle = helix angle of worm.

$\tan \beta = \frac{L}{\pi d}$.

44. Commercial Gears.—The demand for gears is so varied that it is not feasible for the manufacturers to attempt to carry in stock gears to meet all conditions. This is especially true for other than cast iron spur gears with cut teeth. Such gears, completely finished, are carried in stock by some makers in several of the more commonly used pitches and with enough different numbers of teeth to meet most needs. The pitches most commonly carried are 8, 12, 16, 20 and 24. The numbers of teeth vary by somewhat irregular intervals from 12 up to 150 or more. Considerably fewer sizes of cast iron bevel gears with cut teeth, in pairs, for shafts at 90° , are also carried in stock. These include mitre gears in numerous pitches from 4 to 32 and gears with velocity ratios from 3:2 to 4:1 in three or four different pitches. Finished spur and bevel gears with cast teeth are carried in somewhat larger pitches and less variety of numbers of teeth. Very small spur and bevel gears and worm wheels of brass are also carried. While this statement gives a general idea of what may be obtained exact information must be secured from the catalogs of the individual makers. Wherever it is feasible to use these finished stock gears a consider-

able saving in expense is effected. No changes from stock dimensions may be made, however, except as to bore of hub and that at an added cost.

Patterns for spur gears of any desired pitch and number of teeth, and for pairs of bevel gears of any pitch and velocity ratio, are usually carried in stock. Gears from these patterns may be quickly cast to order. Finished patterns for spur and bevel mortise gears are carried in stock by some manufacturers in somewhat fewer sizes.

45. Cams.—Cams are not carried in stock commercially. The sizes in Table XVI are given by Guldner for gas engine cams having

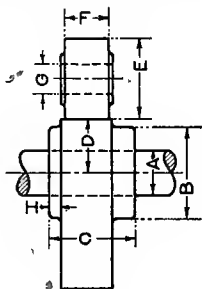


FIG. 58.

hardened steel rolls, carrying a load not to exceed 3000 pounds per inch of length. In the absence of other proportions these may be used for disk cams in general within the range of sizes given. When the diameter of cam shaft is 1" or less the diameter of roll may be made $1\frac{1}{2}$ times the diameter of shaft.

The other dimensions shown in the key drawing, Fig. 58, may be taken as given by the equations below. All dimensions are in inches.

$$B = \text{diameter of hub or boss} = 1\frac{3}{4}A + \frac{1}{4}''.$$

$$C = \text{length of hub} = A \text{ to } 2A.$$

$$D = \text{least radius of cam or radius of base circle} \\ = \frac{1}{2}B + (\frac{1}{16}'' \text{ to } \frac{1}{8}''). \text{ This may be made larger if conditions require a larger cam.}$$

H may be made to suit or may be omitted entirely. The face of the cam is usually made a little wider than the length of the roll. The roll is preferably carried in a forked end, but it may be offset if necessary to save space.

TABLE XVI.
GAS ENGINE CAMS AND ROLLS.

Diameter of Shaft.	Diameter of Roll.	Length of Roll.		Diameter of Pin.
		Minimum.	Maximum.	
A	E	F	F	G
$1\frac{1}{4}$	$1\frac{5}{8}$	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{9}{16}$
$1\frac{1}{2}$	$1\frac{3}{4}$	$\frac{3}{8}$	$\frac{3}{4}$	$\frac{5}{8}$
$1\frac{3}{4}$	$2\frac{1}{4}$	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{3}{4}$
$1\frac{1}{2}$	$2\frac{1}{2}$	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{3}{4}$
$1\frac{1}{2}$	$2\frac{3}{4}$	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{3}{4}$
$1\frac{1}{2}$	$3\frac{1}{4}$	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{3}{4}$
2	$3\frac{1}{2}$	$\frac{1}{2}$	$1\frac{1}{8}$	1
$2\frac{1}{4}$	4	$1\frac{1}{8}$	$1\frac{1}{8}$	$1\frac{3}{16}$
$2\frac{1}{2}$		$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$

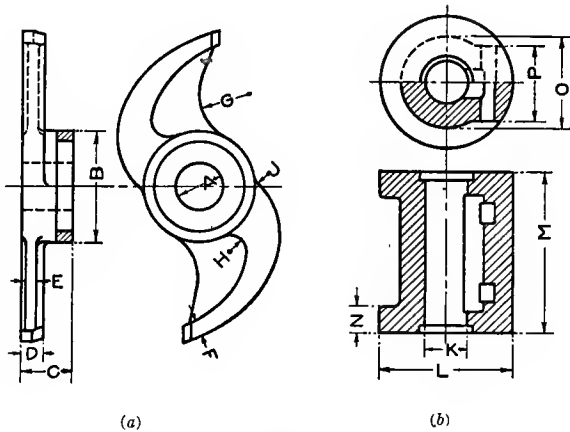


FIG. 59.

The following equations give good proportions for the stamp mill cam and its tappet shown in Fig. 59(a) and (b).

A = diameter of cam shaft.

$$B = 2\frac{3}{8} A.$$

$$C = 1\frac{1}{8} A.$$

$$D = \frac{1}{2} A.$$

$$E = \frac{1}{2} D.$$

$$F = 1\frac{1}{2} E.$$

$$G = A.$$

$$H = \frac{1}{4} A.$$

$$J = \frac{5}{8} A.$$

K = diameter of stamp stem.

$$L = 3 K.$$

$$M = 3\frac{1}{2} K.$$

$$N = \frac{5}{8} K.$$

$$O = 2 K.$$

$$P = 1\frac{2}{3} K.$$

CHAPTER VII.

PIPE AND PIPE FITTINGS.

46. Varieties of Pipe.—Pipe, made from various materials such as wood, tile, cast iron, wrought iron, steel, lead, brass, etc., is kept in stock by manufacturers. The processes by which each of these is formed vary widely. They include casting in a mold; squirting through a hole partially closed by a mandrel; forcing a mandrel through a solid billet of metal; bending and welding the edges of a strip of metal; and bending and riveting the edges of a sheet of metal. The discussion in this book will be confined to wrought iron and steel pipe formed by bending and welding.

47. Wrought Iron and Steel Pipe.—Pipe made from steel has very largely replaced wrought iron pipe for the conveying of water, gas and steam. Wrought iron remains in use chiefly in those places where resistance to corrosion is a determining factor. The terms "wrought iron" and "steel" are used so indiscriminately in this connection that a positive statement is necessary wherever either would not be acceptable. Steel is usually supplied when not otherwise specified. The methods of manufacturing these two are very similar. The metal is rolled into strips of the correct thickness for the pipe and of a width which will bend into a tube with sufficient allowance for welding. The smaller diameters of pipe are butt welded while larger diameters are lap welded. The ends are trimmed back as far as necessary to remove all imperfectly welded portions and, on standard pipe 12" and less in diameter, threaded. An additional charge is made for threading extra weight or large diameter pipe.

Four thicknesses or weights of pipe 12" and less in diameter are made. These are commercially known as **merchant, card or full weight, extra strong and double extra strong** pipe. The term "standard" is applied by some to full weight pipe and by others to merchant pipe. Merchant pipe is generally understood unless otherwise specified but great care must be taken when using the term. The inside diameter of the card or full weight pipe roughly

approximates the nominal size. The outside diameter for any nominal size is made the same for each weight so that the same fittings and threading machinery may be used. The proportions for the commercial sizes of wrought iron and steel pipe and the standard threading for each are given in Table XVII. The proportions for merchant pipe differ so little from those for full weight pipe that no different values are listed by the manufacturers. The

TABLE XVII.
WROUGHT IRON AND STEEL PIPE AND PIPE THREADS.

Nominal.	DIAMETERS.				AREAS.			THREADS.		
	Actual Outside = D.	Actual Inside.			Inside.			Number per Inch = N.	Minimum Root Diameter.	Length of Perfect Thread. Normal Distance Pipe Enters Fitting = L.
		Full Weight.	Extra Strong.	Double Extra Strong.	Full Weight.	Extra Strong.	Double Extra Strong.			
	0.405	0.270	0.205		0.057	0.033		27	0.334	0.19
	0.540	0.364	0.294		0.104	0.068		18	0.433	0.29
	0.675	0.494	0.421		0.191	0.139		18	0.568	0.30
	0.840	0.623	0.542	0.244	0.304	0.231	0.047	14	0.701	0.39
	1.050	0.824	0.736	0.422	0.533	0.425	0.140	14	0.911	0.40
1	1.315	1.048	0.951	0.587	0.861	0.710	0.271	11½	1.144	0.51
1½	1.660	1.380	1.272	0.885	1.496	1.271	0.615	11	1.488	0.54
2	1.900	1.610	1.494	1.088	2.038	1.753	0.930	11	1.728	0.55
2½	2.375	2.067	1.933	1.491	3.356	2.935	1.744	11	2.201	0.58
3	2.875	2.468	2.315	1.755	4.780	4.209	2.419	8	2.619	0.89
3½	3.500	3.067	2.892	2.284	7.388	6.569	4.097	8	3.241	0.95
4	4.000	3.548	3.358	2.716	9.887	8.856	5.794	8	3.738	1.00
4½	4.500	4.026	3.818	3.136	12.730	11.449	7.724	8	4.234	1.05
5	5.000	4.508	4.280	3.564	15.961	14.387	9.976	8	4.731	1.10
	5.562	5.045	4.813	4.063	19.990	18.193	12.965	8	5.290	1.16
6	6.625	6.065	5.751	4.875	28.886	25.976	18.665	8	6.346	1.26
7	7.625	7.023	6.625	5.875	38.743	34.472	27.109	8	7.340	1.36
8	8.625	7.982	7.625	6.875	50.021	45.664	37.122	8	8.334	1.46
9	9.625	8.937	8.625		62.722	58.426		8	9.327	1.57
10	10.750	10.019	9.750		78.822	74.662		8	10.445	1.68
11	11.750	11.000	10.750		95.034	90.764		8	11.439	1.78
12	12.750	12.000	11.750		113.098	108.430		8	12.433	1.88
14	14.000	13.250			137.887			8	13.675	2.00

mill lengths usually approximate to 24 feet but may include pieces as short as 12 feet. A small extra charge is made for pipe cut to specified lengths. This, however, may be more economical when the desired lengths are known, than to cut the pipe on the job. Specify nominal size, weight and length.

In pipe 14" and more in diameter the nominal size is the outside diameter. This is commonly known as **outside diameter** or **O. D.** pipe. The stock sizes vary by 1" up to 22" and by 2" from 22" to 30". Such pipe is made in thicknesses varying by $\frac{1}{16}$ " from $\frac{1}{4}$ " to $\frac{3}{4}$ " for sizes up to 20", from $\frac{5}{16}$ " to $\frac{3}{4}$ " up to 22", from $\frac{3}{8}$ " to $\frac{3}{4}$ " up to 28" and from $\frac{7}{16}$ " to $\frac{3}{4}$ " for 30", except that a thickness of $\frac{1}{16}$ " is not made in any of the diameters. $\frac{3}{8}$ " and $\frac{1}{2}$ " are the more generally used thicknesses. Outside diameter pipe is generally furnished with plain ends and random lengths. Mill lengths run about 24 feet. Specify nominal size, thickness and length.

48. Pipe Threads.—Upon the recommendation of a committee of the American Society of Mechanical Engineers the Briggs standard of pipe threads was adopted on October 27, 1886, by the manufacturers in the United States of wrought iron pipe for gas,

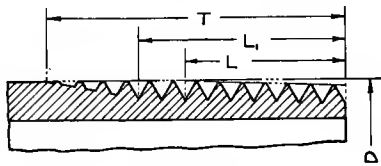


FIG. 60.

steam and water. The change to steel pipe has caused no change in the standard used. A cross section through the Briggs threads appears in the longitudinal section of pipe shown in Fig. 60. The sides of the thread section make an angle of 60° with each other and with the axis of the pipe. The top of the thread and the bottom of the space in the theoretical form are rounded slightly making the depth 0.8 of the pitch. In general this rounding of the corners of the thread is not obtained in practice owing to difficulty in grinding the cutting tools. A sharp V form at the root, and the point cut off flat, leaving the depth 0.833 of the pitch, is the usual section found. The thread is cut on a taper of $\frac{3}{4}$ " per foot. In

addition to the threads perfect at both top and bottom there are two effective threads imperfect at the top only. Normally the pipe should enter the fitting a distance equal to the length of perfect thread. The two additional turns provide a margin in case of imperfect fitting. The remainder of the thread is not of full depth and does not enter the fitting. The following equations give the complete proportions for the thread:

D = outside diameter of pipe;

N = number of threads per inch;

L = length of perfect thread = $\frac{4.8'' + 0.8 D}{N}$;

L_1 = length of effective thread = $\frac{6.8'' + 0.8 D}{N}$;

T = total threaded length = $\frac{10.8'' + 0.8 D}{N}$.

The values of D, N and L for the commercial sizes of pipe are given in Table XVII on page 80.

49. Pipe Joints.—There are two general types of pipe joints. The adjacent ends of two sections of pipe may be screwed into a simple threaded sleeve, called a **coupling**, or **flanges** may be attached to the ends of the sections of pipe and the adjacent flanges bolted together. The former of these types is very much used for pipe of small diameters under low pressures.

50. Couplings.—These couplings are made of wrought iron with right-hand threads for all standard sizes of pipe up to 12" and are furnished with the pipe unless it is ordered flanged. For 2" pipe and smaller, couplings of malleable iron with right-hand threads are carried in stock. For 3" pipe and smaller, couplings of malleable iron threaded one end right-hand and the other end left-hand are carried in stock. For all couplings the nominal size is the same as the nominal size of the pipe which it fits. They are carried in stock either black or galvanized. Black is usually supplied unless otherwise specified.

51. Pipe Flanges.—The most common type of pipe flange is that shown (with a section cut away) in Fig. 61. These flanges screw on the ends of the pipe. Thin rings of rubber, asbestos, or

some soft metal like copper or lead, are placed between the flanges before bolting together, in order to make the joint tight. These rings are called **gaskets**. The pipe should extend sufficiently through the flange to be faced off even with the face of the flange



FIG. 61.

and should bear on the gasket. The bolts are used in multiples of four and are equally spaced to straddle the center lines. The bolt holes are drilled $\frac{1}{8}$ " larger than the nominal diameter of the bolts.

Formerly there were two recognized systems of pipe flanges for different pressures. These were the A. S. M. E. Standard Pipe Flanges for pressures up to 125 pounds per square inch and the

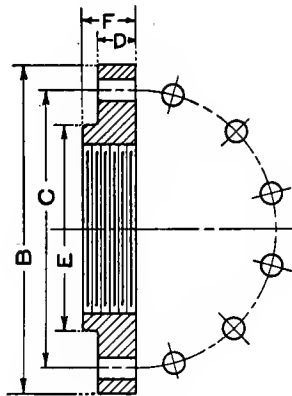


FIG. 62.

Manufacturers' Standard Pipe Flanges for pressures up to 250 pounds per square inch. Both of these systems are being superseded by the **American or United States Standard Pipe Flanges** for each of the above pressures. These two weights of flanges are known respectively as **standard** and **extra heavy**. Values for the dimensions indicated in Fig. 62 have been determined for nominal diameters of pipe up to 100" for standard flanges, and up to

48" for extra heavy flanges. The values of these dimensions and the numbers and sizes of bolts for each weight for sizes up to 48" are given in Table XVIII. When the diameter of the bolt is $1\frac{5}{8}$ " or over stud bolts are recommended. Bolts with square heads and hexagon nuts are recommended for smaller sizes. Standard flanges 32" and over and all extra heavy flanges are spot bored for nuts. On all extra heavy flanges the finished face extends out to within $\frac{1}{32}$ " from the inside edge of the bolt holes. The thickness of the flanges outside this point is reduced $\frac{1}{16}$ " in order that the full pull of the bolts may be exerted in compressing the gasket between the finished faces. Devices to keep the gasket from being blown out, such as a circular depression in one flange in which the gasket is placed and compressed by means of a cor-



FIG. 63.

responding projection on the mating flange, are not mentioned in the specifications of the American Standard. They are, however, much used in high pressure work.

The screwed joint between the pipe and flange, while the most common, is not satisfactorily effective under high pressures. Numerous methods have been devised to improve it. The one which, for cast flanges, seems to have proved most generally satisfactory is shown in Fig. 63. The pipe is passed entirely through the flange and the end afterward expanded to form a flange of its own. It is this latter flange which is faced true to form the seat for the gasket. The cast iron flanges serve as bolting rings to draw the sections of pipe together. Flanges made of steel may be welded to the ends of the pipe, but this method is proportionately more expensive. Any of these methods are open to the objection that the faces must be machined after the flange is on. They cannot, therefore, be used in many places where machinery is not available. This objection is not very serious since, for the

TABLE XVIII.
AMERICAN STANDARD PIPE FLANGES.

Size of Pipe.	Standard—125 Lbs. Pressure.					Extra Heavy—250 Lbs. Pressure.				
	Flange.			Bolts.		Flange.			Bolts.	
	Diameter.			Number.	Diameter.	Diameter.			Number.	Diameter.
	Outside.	Bolt Circle.	Thickness.			Outside.	Bolt Circle.	Thickness.		
	B	C	D			B	C	D		
1	4	3	$\frac{7}{16}$	4	$\frac{7}{16}$	4 $\frac{1}{2}$	3 $\frac{1}{2}$	$\frac{11}{16}$	4	$\frac{1}{2}$
1 $\frac{1}{4}$	4 $\frac{1}{2}$	3 $\frac{3}{8}$	$\frac{7}{16}$	4	$\frac{7}{16}$	5	3 $\frac{3}{4}$	$\frac{11}{16}$	4	$\frac{1}{2}$
1 $\frac{1}{2}$	5	3 $\frac{5}{8}$	$\frac{7}{16}$	4	$\frac{7}{16}$	6	4 $\frac{1}{2}$	$\frac{11}{16}$	4	$\frac{1}{2}$
2	6	4 $\frac{1}{2}$	$\frac{7}{16}$	4	$\frac{7}{16}$	6 $\frac{1}{2}$	5	$\frac{7}{8}$	4	$\frac{1}{2}$
2 $\frac{1}{2}$	7	5 $\frac{1}{2}$	$\frac{7}{16}$	4	$\frac{7}{16}$	7 $\frac{1}{2}$	5 $\frac{7}{8}$	1	4	$\frac{1}{2}$
3	7 $\frac{1}{2}$	6	$\frac{3}{4}$	4	$\frac{5}{8}$	8 $\frac{1}{4}$	6 $\frac{5}{8}$	1 $\frac{1}{8}$	8	$\frac{3}{4}$
3 $\frac{1}{2}$	8 $\frac{1}{2}$	7	$\frac{3}{4}$	4	$\frac{5}{8}$	9	7 $\frac{1}{2}$	1 $\frac{3}{16}$	8	$\frac{3}{4}$
4	9	7 $\frac{1}{2}$	$\frac{3}{4}$	8	$\frac{5}{8}$	10	7 $\frac{3}{4}$	1 $\frac{1}{2}$	8	$\frac{3}{4}$
4 $\frac{1}{2}$	9 $\frac{1}{4}$	7 $\frac{3}{4}$	$\frac{3}{4}$	8	$\frac{5}{8}$	10 $\frac{1}{2}$	8 $\frac{1}{2}$	1 $\frac{5}{8}$	8	$\frac{3}{4}$
5	10	8 $\frac{1}{2}$	$\frac{3}{4}$	8	$\frac{5}{8}$	11	9 $\frac{1}{4}$	1 $\frac{5}{8}$	8	$\frac{3}{4}$
6	11	9 $\frac{1}{2}$	1	8	$\frac{3}{4}$	12 $\frac{1}{2}$	10 $\frac{3}{8}$	1 $\frac{7}{16}$	12	$\frac{3}{4}$
7	12 $\frac{1}{2}$	10 $\frac{3}{4}$	1 $\frac{1}{16}$	8	$\frac{3}{4}$	14	11 $\frac{7}{8}$	1 $\frac{1}{2}$	12	$\frac{3}{4}$
8	13 $\frac{1}{2}$	11 $\frac{1}{2}$	1 $\frac{1}{8}$	8	$\frac{3}{4}$	15	13	1 $\frac{3}{8}$	12	$\frac{3}{4}$
9	15	13 $\frac{1}{4}$	1 $\frac{1}{8}$	12	$\frac{3}{4}$	16 $\frac{1}{2}$	14	1 $\frac{3}{4}$	12	1
10	16	14 $\frac{1}{4}$	1 $\frac{1}{8}$	12	$\frac{3}{4}$	17 $\frac{1}{2}$	15 $\frac{1}{4}$	1	16	1
12	19	17	1 $\frac{1}{4}$	12	$\frac{7}{8}$	20 $\frac{1}{2}$	17 $\frac{3}{4}$	2	16	1 $\frac{1}{8}$
14	21	18 $\frac{3}{4}$	1 $\frac{3}{8}$	12	1	23	20 $\frac{1}{2}$	2 $\frac{1}{8}$	20	1 $\frac{1}{8}$
15	22 $\frac{1}{4}$	20	1 $\frac{3}{8}$	16	1	24 $\frac{1}{2}$	21 $\frac{1}{2}$	2 $\frac{3}{16}$	20	1 $\frac{1}{4}$
16	23 $\frac{1}{2}$	21 $\frac{1}{2}$	1 $\frac{7}{8}$	16	1	25 $\frac{1}{2}$	22 $\frac{1}{2}$	2 $\frac{1}{4}$	20	1 $\frac{1}{4}$
18	25	22 $\frac{1}{4}$	1 $\frac{1}{8}$	16	1 $\frac{1}{8}$	28	24 $\frac{1}{4}$	2 $\frac{3}{8}$	24	1 $\frac{1}{4}$
20	27 $\frac{1}{2}$	25	1 $\frac{11}{16}$	20	1 $\frac{1}{8}$	30 $\frac{1}{2}$	27	2 $\frac{1}{2}$	24	1 $\frac{3}{8}$
22	29 $\frac{1}{2}$	27 $\frac{1}{4}$	1 $\frac{13}{16}$	20	1 $\frac{1}{4}$	33	29 $\frac{1}{4}$	2 $\frac{5}{8}$	24	1 $\frac{1}{2}$
24	32	29 $\frac{1}{2}$	1 $\frac{7}{8}$	20	1 $\frac{1}{4}$	36	32	2 $\frac{3}{4}$	24	1 $\frac{5}{8}$
26	34 $\frac{1}{4}$	31 $\frac{3}{4}$	2	24	1 $\frac{1}{4}$	38 $\frac{1}{4}$	34 $\frac{1}{2}$	2 $\frac{11}{16}$	28	1 $\frac{5}{8}$
28	36 $\frac{1}{2}$	34	2 $\frac{1}{16}$	28	1 $\frac{1}{4}$	40 $\frac{5}{4}$	37	2 $\frac{1}{8}$	28	1 $\frac{5}{8}$
30	38 $\frac{3}{4}$	36	2 $\frac{1}{8}$	28	1 $\frac{3}{8}$	43	39 $\frac{1}{4}$	3	28	1 $\frac{3}{4}$
32	41 $\frac{1}{2}$	38 $\frac{1}{2}$	2 $\frac{1}{4}$	28	1 $\frac{1}{2}$	45 $\frac{1}{4}$	41 $\frac{1}{2}$	3 $\frac{1}{8}$	28	1 $\frac{7}{8}$
34	43 $\frac{3}{4}$	40 $\frac{1}{2}$	2 $\frac{5}{16}$	32	1 $\frac{1}{2}$	47 $\frac{1}{2}$	43 $\frac{1}{2}$	3 $\frac{1}{4}$	28	1 $\frac{7}{8}$
36	46	42 $\frac{3}{4}$	2 $\frac{3}{8}$	32	1 $\frac{1}{2}$	50	46	3 $\frac{3}{8}$	32	1 $\frac{7}{8}$
38	48 $\frac{3}{4}$	45 $\frac{1}{4}$	2 $\frac{3}{8}$	32	1 $\frac{5}{8}$	52 $\frac{1}{4}$	48	3 $\frac{1}{16}$	32	1 $\frac{7}{8}$
40	50 $\frac{3}{4}$	47 $\frac{1}{4}$	2 $\frac{1}{2}$	36	1 $\frac{5}{8}$	54 $\frac{1}{2}$	50 $\frac{1}{4}$	3 $\frac{9}{16}$	36	1 $\frac{7}{8}$
42	53	49 $\frac{3}{4}$	2 $\frac{5}{8}$	36	1 $\frac{5}{8}$	57	52 $\frac{3}{4}$	3 $\frac{1}{4}$	36	1 $\frac{7}{8}$
44	55 $\frac{1}{4}$	51 $\frac{3}{4}$	2 $\frac{5}{8}$	40	1 $\frac{5}{8}$	59 $\frac{1}{2}$	55	3 $\frac{3}{4}$	36	2
46	57 $\frac{1}{4}$	53 $\frac{1}{4}$	2 $\frac{1}{8}$	40	1 $\frac{5}{8}$	61 $\frac{1}{2}$	57 $\frac{1}{4}$	3 $\frac{7}{8}$	40	2
48	59 $\frac{1}{2}$	56	2 $\frac{3}{4}$	44	1 $\frac{5}{8}$	65	60 $\frac{3}{4}$	4	40	2

best results, the screwed flange should also be trued up after it is on. Except on large contracts, where it would pay to set up the necessary machinery on the job, it is advisable to have the pipe cut to lengths and all flanges put on and faced true at the mill.

52. **Pipe Bends.**—Besides the straight sections, pipe in sizes up to 24" may be obtained curved to various forms such as those shown in Fig. 64. The curved portions are bent to the arc of a circle but short straight portions at the ends are necessary in mak-

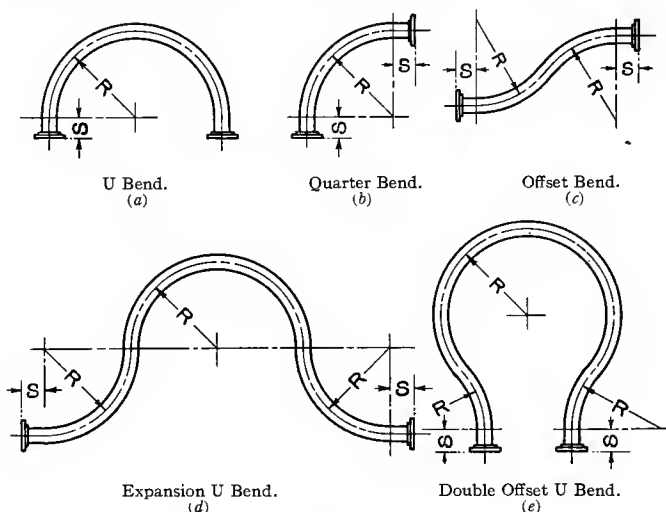


FIG. 64.

ing the bends. The recommended values for the radii of the curved portions and the necessary lengths for the straight ends are given in Table XIX. By using extra strong pipe these radii may be reduced to the minimum values given in the table but the makers do not guarantee such bends against buckling. Pipe bends are usually made to order. The distance between the flange faces and the amount of the offset in the **offset bend**, Fig. 64(c), and the amount of the offset in the **double offset U bend**, Fig. 64(e), are optional within certain limits. Straight portions of limited length may usually be added where desired in any of the forms. In all cases pipe bends are furnished with the flanges on and faced true to the desired angle.

TABLE XIX.
PROPORTIONS FOR PIPE BENDS.

Size of Pipe.	Radius of Bend.		Straight Ends.	Size of Pipe.	Radius of Bend.		Straight Ends.
	Standard.	Minimum.			Standard.	Minimum.	
	R	R			R	R	
2½	12½	7	4	10	50	40	12
3	15	8	4	12	60	50	14
3½	17½	10	5	14	70	65	16
4	20	12	5	15	75	70	16
4½	22½	14	6	16	80	78	18
5	25	15	6	18	108	88	18
6	30	20	7	20	120	104	18
7	35	24	8	22	132	132	18
8	40	28	9	24	144	144	18
9	45	35	11				

53. Pipe Fittings.—The forms of pipe fittings are too numerous for complete description or illustration in this book. They are usually of cast iron but steel or bronze castings may be required for special work. Their flanges conform to the specifications given for pipe flanges in article 51. The finished center to face or face to face distances and the minimum thicknesses of body metal in standard fittings for sizes of pipe up to 48" are given in Tables XX and XXI and corresponding values for extra heavy fittings are given in Tables XXII and XXIII. The symbols apply to the key drawings, Fig. 65 on page 91, which illustrate some of the more common forms. The nominal size of pipe is that corresponding to the largest opening in the fitting.

The direct passage through a fitting is known as the **run**. The side openings are called **branches**. When these openings are of different sizes those of the run are given first and followed by those of the branches. Tees, crosses and laterals are made in two lengths of run, known respectively as **short body** and **long body** patterns. The short body is used only when the diameter of the largest branch is considerably less than (approximately two-thirds) that of the run, in sizes of pipe 18" or larger. The maximum diameter of branch for the short body pattern and the minimum outlets for these fittings are given in the tables.

PIPE AND PIPE FITTINGS

TABLE XX.

AMERICAN STANDARD FLANGED FITTINGS.

Tees, Crosses and Elbows. Standard Weight—125 Lbs. Pressure.									
Size of Pipe.	Maximum Branch for Short Body Pattern.	Minimum Outlet.	Distance Center to Face.						Minimum Thickness of Metal in Body.
			Tees and Crosses.			Elbows.			
			Long Body.	Short Body.		90°.	45°.	Long Radius.	
				Run.	Branch.				
			A	B	C	A	D	E	
1		1	3½			3½	1¾	5	7⁄8
1¼		1	3¾			3¾	2	5½	7⁄8
1½		1	4			4	2¼	6	7⁄8
2		1	4½			4½	2½	6½	7⁄8
2½		1	5			5	3	7	7⁄8
3		1	5½			5½	3	7¾	7⁄8
3½		1	6			6	3½	8½	7⁄8
4		1	6½			6½	4	9	7⁄8
4½		1	7			7	4	9½	7⁄8
5		1	7½			7½	4½	10¼	7⁄8
6		2	8			8	5	11½	9⁄8
7		2	8½			8½	5½	12¾	9⁄8
8		2	9			9	5½	14	9⁄8
9		2	10			10	6	15¼	9⁄8
10		2	11			11	6½	16½	9⁄8
12		2	12			12	7½	19	11⁄8
14		3	14			14	7½	21½	11⁄8
15		3	14½			14½	8	22¾	11⁄8
16		3	15			15	8	24	11⁄8
18	12	3	16½	13	15½	16½	8½	26½	11⁄8
20	14	3	18	14	17	18	9½	29	11⁄8
22	15	3	20	14	18	20	10	31½	11⁄8
24	16	3	22	15	19	22	11	34	11⁄8
26	18	3	23	16	20	23	13	36½	11⁄8
28	18	3	24	16	21	24	14	39	11⁄8
30	20	3	25	18	23	25	15	41½	11⁄8
32	20	3	26	18	24	26	16	44	11⁄8
34	22	3	27	19	25	27	17	46½	11⁄8
36	24	3	28	20	26	28	18	49	11⁄8
38	24	3	29	20	28	29	19	51½	11⁄8
40	26	3	30	22	29	30	20	54	11⁄8
42	28	3	31	23	30	31	21	56½	11⁄8
44	28	3	32	23	31	32	22	59	11⁄8
46	30	3	33	24	33	33	23	61½	11⁄8
48	32	3½	34	26	34	34	24	64	2

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TABLE XXI.

[illegible]

PIPE AND PIPE FITTINGS

 TABLE XXII.
 AMERICAN STANDARD FLANGED FITTINGS.

Tees, Crosses and Elbows. Extra Heavy—250 Lbs. Pressure.										
Size of Pipe.	Maximum Branch for Short Body Pattern.	Minimum Outlet.	Distance Center to Face.						Minimum Thickness of Metal in Body.	
			Tees and Crosses.			Elbows.				
			Long Body.	Short Body.		90°.	45°.	Long Radius.		
				Run.	Branch.					
										A
1		1	4				4		5	$\frac{1}{8}$
$1\frac{1}{4}$		1	$4\frac{1}{4}$				$4\frac{1}{4}$		$5\frac{1}{2}$	$\frac{1}{4}$
$1\frac{1}{2}$		1	$4\frac{1}{2}$				$4\frac{1}{2}$		6	$\frac{1}{4}$
2		1	5				5		$6\frac{1}{2}$	$\frac{3}{8}$
$2\frac{1}{2}$		1	$5\frac{1}{2}$				$5\frac{1}{2}$		7	$\frac{9}{16}$
3		1	6				6		$7\frac{3}{4}$	$\frac{9}{16}$
$3\frac{1}{2}$		1	$6\frac{1}{2}$				$6\frac{1}{2}$		$8\frac{1}{2}$	$\frac{9}{16}$
4		1	7				7		9	$\frac{1}{2}$
$4\frac{1}{2}$		1	$7\frac{1}{2}$				$7\frac{1}{2}$		$9\frac{1}{2}$	$\frac{1}{2}$
5		1	8				8		$10\frac{1}{4}$	$\frac{1}{2}$
6		2	$8\frac{1}{2}$				$8\frac{1}{2}$		$11\frac{1}{2}$	$\frac{3}{4}$
7		2	9				9		$12\frac{3}{4}$	$\frac{1}{2}$
8		2	10				10		14	$\frac{1}{2}$
9		2	$10\frac{1}{2}$				$10\frac{1}{2}$		$15\frac{1}{4}$	$\frac{1}{2}$
10		2	$11\frac{1}{2}$				$11\frac{1}{2}$		$16\frac{1}{2}$	$\frac{1}{2}$
12		2	13				13		19	1
14		2	15				15		$21\frac{1}{2}$	$\frac{1}{2}$
15		2	$15\frac{1}{2}$				$15\frac{1}{2}$		$22\frac{3}{4}$	$\frac{1}{2}$
16		2	$16\frac{1}{2}$				$16\frac{1}{2}$		24	$\frac{1}{2}$
18	12	3	18	14	17		18	10	$26\frac{1}{2}$	$\frac{1}{8}$
20	14	3	$19\frac{1}{2}$	$15\frac{1}{2}$	$18\frac{1}{2}$		$19\frac{1}{2}$	$10\frac{1}{2}$	29	$1\frac{1}{8}$
22	15	3	$20\frac{1}{2}$	$16\frac{1}{2}$	20		$20\frac{1}{2}$	11	$31\frac{1}{2}$	$1\frac{1}{8}$
24	16	3	$22\frac{1}{2}$	17	$21\frac{1}{2}$		$22\frac{1}{2}$	12	34	$1\frac{1}{8}$
26	18	3	24	19	23		24	13	$36\frac{1}{2}$	$1\frac{1}{8}$
28	18	3	26	19	24		26	14	39	$1\frac{1}{8}$
30	20	3	$27\frac{1}{2}$	$20\frac{1}{2}$	$25\frac{1}{2}$		$27\frac{1}{2}$	15	$41\frac{1}{2}$	2
32	20	3	29	$20\frac{1}{2}$	$26\frac{1}{2}$		29	16	44	$2\frac{1}{8}$
34	22	3	$30\frac{1}{2}$	22	28		$30\frac{1}{2}$	17	$46\frac{1}{2}$	$2\frac{1}{8}$
36	24	3	$32\frac{1}{2}$	$23\frac{1}{2}$	$29\frac{1}{2}$		$32\frac{1}{2}$	18	49	$2\frac{1}{8}$
38	24	3	34	$23\frac{1}{2}$	$30\frac{1}{2}$		34	19	$51\frac{1}{2}$	$2\frac{1}{8}$
40	26	3	$35\frac{1}{2}$	25	$31\frac{1}{2}$		$35\frac{1}{2}$	20	54	$2\frac{1}{8}$
42	28	3	37	$26\frac{1}{2}$	$33\frac{1}{2}$		37	21	$56\frac{1}{2}$	$2\frac{1}{8}$
44	28	3	39	$26\frac{1}{2}$	$34\frac{1}{2}$		39	22	59	$2\frac{1}{8}$
46	30	3	$40\frac{1}{2}$	$27\frac{1}{2}$	$35\frac{1}{2}$		$40\frac{1}{2}$	23	$61\frac{1}{2}$	$2\frac{1}{8}$
48	32	$3\frac{1}{2}$	42	29	$37\frac{1}{2}$		42	24	64	3

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[illegible]

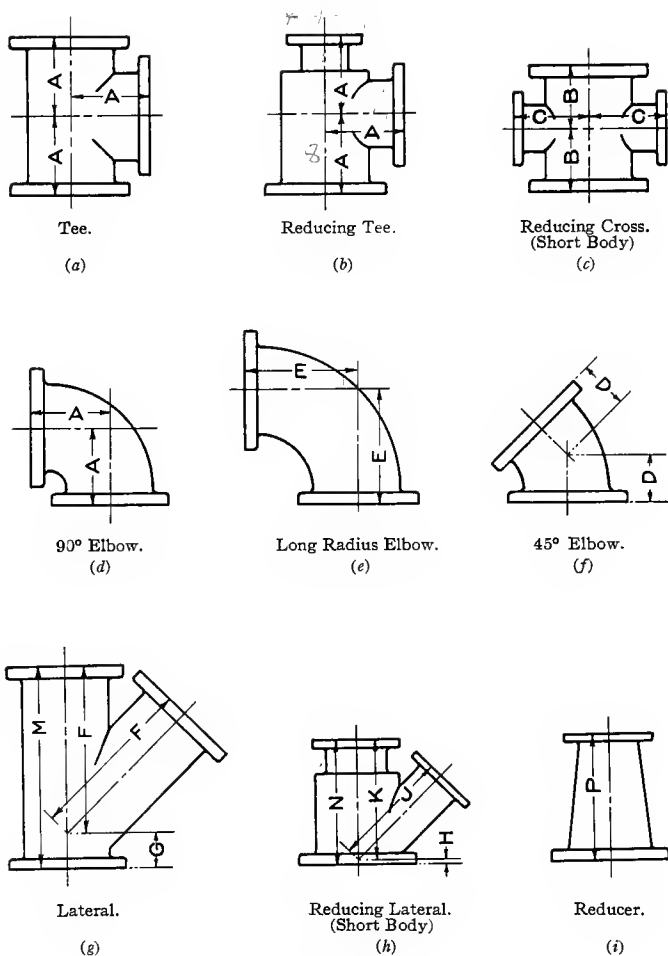


FIG. 65.

54. Valves.—Valves for the purpose of shutting off the flow of fluid through a pipe are made in two general types, namely, globe valves and gate valves. These two types are shown in section in Figs. 66(a) and 66(b) respectively. The gate valve is for some purposes preferred to the globe valve because of the more

direct passage of the fluid when the valve is open. Many other forms of valves are made for special purposes. The openings of

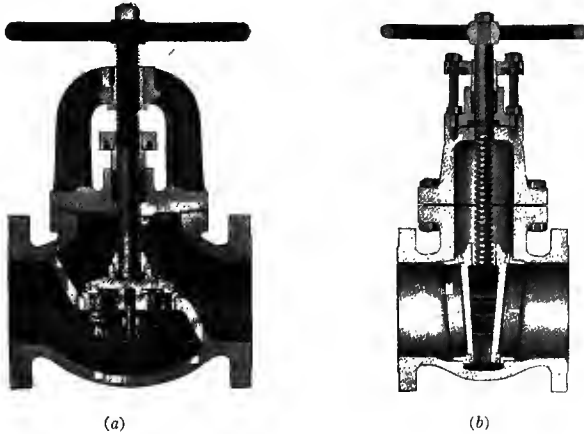


FIG. 66.

the valve may be threaded and the pipe screwed in or they may be flanged. The flanges on valves are made to conform to the specifications given in article 51 on page 82.

DECIMAL EQUIVALENTS OF FRACTIONS OF ONE INCH.

Sixteenths.	Thirty-seconds.	Sixty-fourths.	Decimals.	Sixteenths.	Thirty-seconds.	Sixty-fourths.	Decimals.
16	32	64		16	32	64	
1	1	1	.015625	9	17	33	.515625
		3	.03125			35	.53125
		5	.046875			37	.546875
2	3	7	.0625	10	19	39	.5625
		9	.078125			41	.578125
		11	.09375			43	.59375
3	5	13	.109375	11	23	45	.609375
		15	.125			47	.625
		17	.140625			49	.640625
4	7	19	.15625	12	25	51	.65625
		21	.171875			53	.671875
		23	.1875			55	.6875
5	9	25	.203125	13	27	57	.703125
		27	.21875			59	.71875
		29	.234375			61	.734375
6	11	31	.25	14	29	63	.75
		33	.265625			65	.765625
		35	.28125			67	.78125
7	13	37	.296875	15	31	69	.796875
		39	.3125			71	.8125
		41	.328125			73	.828125
8	15	43	.34375	16	33	75	.84375
		45	.359375			77	.859375
		47	.375			79	.875
		49	.390625			81	.890625
		51	.40625			83	.90625
		53	.421875			85	.921875
		55	.4375			87	.9375
		57	.453125			89	.953125
		59	.46875			91	.96875
		61	.484375			93	.984375
		63	.5			95	
		65				97	

Degrees	TANGENTS							Cotangents
	0'	10'	20'	30'	40'	50'	60'	
0	0.00000	0.00291	0.00582	0.00873	0.01164	0.01455	0.01746	89
1	0.01746	0.02036	0.02327	0.02619	0.02910	0.03201	0.03492	88
2	0.03492	0.03783	0.04075	0.04366	0.04658	0.04949	0.05241	87
3	0.05241	0.05533	0.05824	0.06116	0.06408	0.06700	0.06993	86
4	0.06993	0.07285	0.07578	0.07870	0.08163	0.08456	0.08749	85
5	0.08749	0.09042	0.09335	0.09629	0.09923	0.10216	0.10510	84
6	0.10510	0.10805	0.11099	0.11394	0.11688	0.11983	0.12278	83
7	0.12278	0.12574	0.12869	0.13165	0.13461	0.13758	0.14054	82
8	0.14054	0.14351	0.14648	0.14945	0.15243	0.15540	0.15838	81
9	0.15838	0.16137	0.16435	0.16734	0.17033	0.17333	0.17633	80
10	0.17633	0.17933	0.18233	0.18534	0.18835	0.19136	0.19438	79
11	0.19438	0.19740	0.20042	0.20345	0.20648	0.20952	0.21256	78
12	0.21256	0.21560	0.21864	0.22169	0.22475	0.22781	0.23087	77
13	0.23087	0.23393	0.23700	0.24008	0.24316	0.24624	0.24933	76
14	0.24933	0.25242	0.25552	0.25862	0.26172	0.26483	0.26795	75
15	0.26795	0.27107	0.27419	0.27732	0.28046	0.28360	0.28675	74
16	0.28675	0.28990	0.29305	0.29621	0.29938	0.30255	0.30573	73
17	0.30573	0.30891	0.31210	0.31530	0.31850	0.32171	0.32492	72
18	0.32492	0.32814	0.33136	0.33460	0.33783	0.34108	0.34433	71
19	0.34433	0.34758	0.35085	0.35412	0.35740	0.36068	0.36397	70
20	0.36397	0.36727	0.37057	0.37388	0.37720	0.38053	0.38386	69
21	0.38386	0.38721	0.39055	0.39391	0.39727	0.40065	0.40403	68
22	0.40403	0.40741	0.41081	0.41421	0.41763	0.42105	0.42447	67
23	0.42447	0.42791	0.43136	0.43481	0.43828	0.44175	0.44523	66
24	0.44523	0.44872	0.45222	0.45573	0.45924	0.46277	0.46631	65
25	0.46631	0.46985	0.47341	0.47698	0.48055	0.48414	0.48773	64
26	0.48773	0.49134	0.49495	0.49858	0.50222	0.50587	0.50953	63
27	0.50953	0.51320	0.51688	0.52057	0.52427	0.52798	0.53171	62
28	0.53171	0.53545	0.53920	0.54296	0.54674	0.55051	0.55431	61
29	0.55431	0.55812	0.56194	0.56577	0.56962	0.57348	0.57735	60
30	0.57735	0.58124	0.58513	0.58905	0.59297	0.59691	0.60086	59
31	0.60086	0.60483	0.60881	0.61280	0.61681	0.62083	0.62487	58
32	0.62487	0.62892	0.63299	0.63707	0.64117	0.64528	0.64941	57
33	0.64941	0.65355	0.65771	0.66189	0.66608	0.67028	0.67451	56
34	0.67451	0.67875	0.68301	0.68728	0.69157	0.69588	0.70021	55
35	0.70021	0.70455	0.70891	0.71329	0.71769	0.72211	0.72654	54
36	0.72654	0.73100	0.73547	0.73996	0.74447	0.74900	0.75355	53
37	0.75355	0.75812	0.76272	0.76733	0.77196	0.77661	0.78129	52
38	0.78129	0.78598	0.79070	0.79544	0.80020	0.80498	0.80978	51
39	0.80978	0.81461	0.81946	0.82434	0.82923	0.83415	0.83910	50
40	0.83910	0.84407	0.84906	0.85408	0.85912	0.86419	0.86929	49
41	0.86929	0.87441	0.87955	0.88473	0.88992	0.89515	0.90040	48
42	0.90040	0.90569	0.91099	0.91633	0.92170	0.92709	0.93252	47
43	0.93252	0.93797	0.94345	0.94896	0.95451	0.96008	0.96569	46
44	0.96569	0.97133	0.97700	0.98270	0.98843	0.99420	1.00000	45
Tangents	60'	50'	40'	30'	20'	10'	0'	Degrees
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